DIESEL ENGINE DESIGN

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PREFACE

This book is based on about twelve years' experience of Diesel Engines, mainly from the drawing-office point of view, and is intended to present an account of the main considerations which control the design of these engines.

The author ventures to hope that, in addition to designers and draughtsmen, to whom such a book as this is most naturally addressed, there may be other classes of readers—for example, Diesel Engine users and technical students—to whom the following pages may be of interest.

The text deals mainly with general principles as exemplified by examples of good modern practice, and it has not been possible to notice every constructional novelty. Apology is perhaps called for on account of the omission of any special treatment of the stepped piston and the opposed piston types of engine. These, however, are the specialities of a comparatively limited number of manufacturers, and have been very fully described and illustrated in the technical press.

The existence in its fourth edition of Chalkley's well-known book on *The Diesel Engine for Land and Marine Purposes* has enabled the present writer to proceed to details with a minimum of preliminary discussion. A number of references to other books and papers have been inserted in order to avoid, so far as possible, overlapping with other sources of information.

The author has pleasure in acknowledging his indebtedness to: Mr. P. H. Smith (who has at all times placed his unique experience of Diesel Engines at the disposal of the author) for several corrections and suggestions; to Mr. L. Johnson, M.A. (Cantab.), for his very careful and patient revision of the proofs; to the author's wife for assistance with the manuscript and for compiling the index.

H. F. P. P.

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DIESEL ENGINE DESIGN

CHAPTER I

FIRST PRINCIPLES

The Diesel Principle.—The characteristic feature of the Diesel Engine is the injection of oil fuel into air which has been previously compressed by the rising of a piston to a pressure corresponding to a temperature sufficiently high to ensure immediate ignition of the fuel.

In the course of the pioneer experiments by which the commercial practicability of this engine was demonstrated, it was found advantageous to effect the injection of the fuel by a blast of air, and this feature was retained in all Diesel Engines until the lapse of the original patents.

At the present date there exists a class of high-compression oil engines operating on the Diesel principle, in which the injection of oil is effected by mechanical means without the assistance of an air blast. The design of these engines presents a variety of problems differing materially from those which arise in the design of Diesel Engines as defined below. Furthermore, the use for war purposes of one of the most conspicuous members of this class of engine prohibits anything like a satisfactory discussion of these so-called "solid injection" engines. The well-known "surface ignition," "hot bulb," or "hot plate" engines form a very numerous class by themselves and have in the past been misnamed "semi-Diesel" engines. The cycles on which they operate and the principles underlying their design differ so widely from those relating to Diesel Engines proper that they also fall outside the scope of this work.

The features which characterise the true Diesel Engine, in the correct use of the term, are now understood to be the following :—

(1) Compression sufficient to produce the temperature requisite for spontaneous combustion of the fuel.
(2) Injection of fuel by a blast of compressed air.
(3) A maximum cycle pressure (attained during combustion) not greatly exceeding the compression pressure, i.e. absence of pronounced explosive effect.

Item 3 is deliberately worded somewhat broadly as the shape of a Diesel indicator card is subject to considerable variation under different conditions of load, blast air pressure, fuel valve adjustment, etc.

In the earlier days of Diesel Engine construction the square top indicator card, showing a period of combustion at constant pressure, was considered the ideal to aim at. It has since been found that a card having a more peaked top is usually associated with better fuel consumptions. When tar oil is used as fuel the square top card appears to be almost out of the question.

It should further be remembered that the existence of a period of combustion at constant pressure is no guarantee that all the combustion takes place at that pressure. This ideal is never realised. Combustion probably proceeds slowly well after half stroke, even under the most favourable conditions.

Compression Pressure.—The height to which compression is carried is governed by the following considerations :—

(1) The attainment of the requisite temperature.
(2) The attainment of a desirable degree of efficiency.
(3) Mechanical considerations.

Considerations of temperature for ignition fix the lower limit of compression at somewhere in the neighbourhood of 400 lb. per sq. in. The temperature actually attained depends on the initial temperature of the intaken air and the heat lost to the jacket during compression, so it is clear that the temperature attained on the first few strokes of the engine will be considerably lower than the value it assumes after the engine has been firing consecutively for some time.

As regards efficiency, it is well known that increasing the degree of compression beyond certain limits does not very materially increase even the theoretical efficiency.

In practice the compression most usually adopted is about 500–550 lb. per sq. in. for four stroke engines. For two stroke engines the compression is frequently in the neighbourhood of

600 lb. per sq. in. or over, owing to the fact that the charge of air delivered by the scavenge pump may itself be at a pressure slightly above atmospheric. At first sight it might be thought that this initial compression might be considered as a first stage in the temperature rise of the charge of air; but apparently compression in the scavenge pump is not so effective in raising the temperature as compression in the main cylinder. The mechanical considerations which limit the compression are numerous, and some are mentioned below.

Higher compression involves :—

(1) Heavier load per sq. in. of the piston and necessitates massive construction of all the main parts.

(2) More highly compressed air for injection and consequently increased trouble with the air compressor, and its valves particularly.

(3) Increased wear of cylinder liners due to increased pressure behind the piston rings.

Compression Temperature.—With a compression of 500 lb. per sq. in. in a fair-sized four cycle cylinder working under full load conditions the compression temperature is about 1200° F. On starting the engine from a cold state the compression pressure and temperature are considerably lower owing to the cold state of the cylinder walls and the piston crown.

In addition to this the injection of cold blast air with the fuel in the proportion of about 1 lb. of blast air to 12 lb. of suction air still further reduces the temperature apart from the probability that the blast air has momentarily a local cooling effect in the zone of combustion.

The middle curve (Fig. 1) shows graphically the connection between the compression temperature and compression pressure on the assumptions that :—

(1) The initial temperature of the intaken air is 212° F.

(2) That the exponent in the equation PV^n=const. is 1·35.

These assumptions correspond approximately to the conditions obtaining with a fully loaded engine of fair size—say an 18″ cylinder.

The noteworthy point about this curve is the slowing down of the rate of increase of temperature with pressure as the latter increases. Expressed mathematically $\frac{dT}{dP}$ diminishes as P increases.

The Four Stroke Cycle.—The well-known four stroke cycle consists briefly of :—

(1) The Suction Stroke.
(2) The Compression Stroke.
(3) The Combustion and Expansion Stroke.
(4) The Exhaust Stroke.

These are considered in detail below.

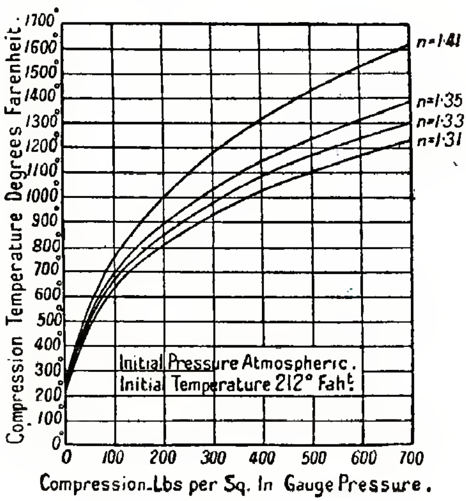


FIG. 1.

Suction Stroke.—If the engine crank is considered to be at its inner dead centre and just about to begin the suction stroke, the suction valve is already slightly open. In steam engine parlance it has a slight lead. At the same time the exhaust valve, which has been previously closing on the exhaust stroke, has not yet come on its seat. The result of this state of affairs is that the rapidly moving exhaust gases create a partial vacuum in the combustion space and induce a flow of air through the suction valve, thus tending to scavenge out exhaust gases which would otherwise remain in the cylinder.

As the piston descends its velocity increases and reaches a maximum in the neighbourhood of half stroke. At the same time the suction valve is being lifted further off its seat and

attains its maximum opening also in the neighbourhood of half stroke. The lower half of the suction stroke is accompanied by a more or less gradual closing of the suction valve, which, however, is not allowed to come on its seat until the crank has passed the lower dead centre by about 20°. At the moment when the crank is passing the lower dead centre the induced air is passing through the restricted opening of the rapidly closing suction valve with considerable velocity and an appreciable duration of time must elapse before the upward movement of the piston can effect a reversal of the direction of flow through the suction valve. It will be clear from the above that owing to the effect of inertia more air will be taken into the cylinder in the manner described than by allowing the suction valve to come on its seat exactly at the bottom dead centre. The exact point at which the suction valve should close is doubtless capable of approximate calculation, but is usually fixed in accordance with current practice or test-bed experiments.

Compression Stroke.—The piston now rises on its up stroke and compresses the air to about 500 lb. per sq. in. The clearance volume necessary for this compression being about 8% of the stroke volume. During the compression the temperature rises and a certain amount of heat is lost to the cylinder walls and cylinder cover. The final compression temperature is in the neighbourhood of 1200° F.

Combustion and Expansion Stroke.—At the upper dead centre, or slightly previous thereto, the injection valve opens and fuel oil is driven into the cylinder and starts burning immediately. The actual point at which the fuel enters the cylinder is not quite certain, as there is inevitably some lag between the opening of the injection valve and the entrance of fuel. The point at which the fuel valve starts to open, as determined by a method described below, varies from about 3° (slow speed engines) to 14° (high speed engines). The method of determining the point of opening of the fuel valve is as follows :—

With the engine at rest, air at about 100 lb. pressure is turned on to the injection valve and then communication with the blast air-bottle is cut off to prevent unnecessary waste of air and the possibility of the engine turning under the impulse of the air which is subsequently admitted to the cylinder. The indicator cock is now opened and the engine slowly barred

round by hand until the air is heard to enter the cylinder by placing the ear to the indicator cock. The position of the engine when this occurs is the nearest possible approximation to the true point of opening, assuming the operation has been carefully done.

The duration of the fuel valve opening is usually about 48°, and in the majority of engines is fixed for all loads. It is evident that at light load the opening is longer than necessary, and in some designs the duration of opening is regulated by the governor in accordance with the load.

The combustion is by no means complete when the fuel valve closes, and usually continues in some measure well past the half stroke of the engine. This is known as "after burning,"

FIG. 2.

and takes place with the very best engines in the best state of adjustment. Exaggerated after burning is the surest sign of misadjustment, and makes itself apparent by abnormally high terminal pressure at the point at which the exhaust valve opens, and is readily detected on an indicator card by comparison with that taken from an engine in good adjustment. As will be shown later, the presence of "after burning" is most clearly seen on an Entropy Diagram.

Expansion continues accompanied by loss of heat to the cylinder walls until the exhaust valve opens.

Exhaust Stroke.—The exhaust valve opens about 50° before the bottom dead centre, in order that the exhaust gases may effect a rapid escape and reduce the back pressure on the exhaust stroke. The pressure in the cylinder when the exhaust valve starts to open is about 40 lb. per sq. in. with an engine working with a mean indicated pressure of 100 lb. per sq. in. The temperature of the exhaust gases at this point is some-

where in the neighbourhood of about 1600° F. and the velocity is consequently very high.

The pressure falls nearly to atmospheric shortly after the bottom dead centre has been passed and the back pressure during the remainder of the exhaust stroke should not be more than about 1 lb. per sq. in., or less. Excessive back pressure may arise from :—

(1) Insufficient diameter or lift or late opening of exhaust valve.
(2) Exhaust pipe too small in diameter.
(3) Obstructions or sharp bends in the exhaust pipe or silencer.
(4) Interference by another cylinder exhausting into the same pipe.

It is interesting to note that owing to the higher velocity of air at high temperature per unit pressure difference the back pressure is more at light load than at full load.

Indicator Cards.—Figs. 2 and 3 show typical indicator cards taken with a heavy and a light spring respectively.

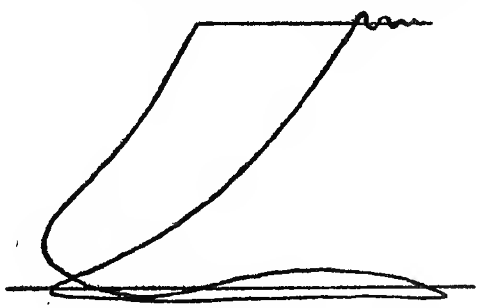


FIG. 3.

The latter is particularly useful for investigating the processes of suction and exhaust.

It is to be observed that in Fig. 3 the compression is seen to start at a point which is indistinguishable from the bottom dead centre, thus indicating a volumetric efficiency of practically 100%. This is to be regarded as a normal state of affairs, obtainable with both high speed and low speed engines. The volumetric efficiencies of internal combustion engines are frequently quoted at figures varying between about 95% for slow speed engines to 80% for high speed engines. The former

figure is reasonable, but the latter can only be due either to imperfect design (or adjustment) of the engine or to erroneous indicator cards. The use of too weak a spring in the indicator may lead to a diagram showing not more than 60% volumetric efficiency, owing to the inertia of the indicator piston, etc. Consequently, fairly stiff springs are to be preferred.

Valve Setting Diagram.—Fig. 4 is a typical valve setting diagram for a four stroke engine, and shows the points relative to the dead centres at which the various valves open and close.

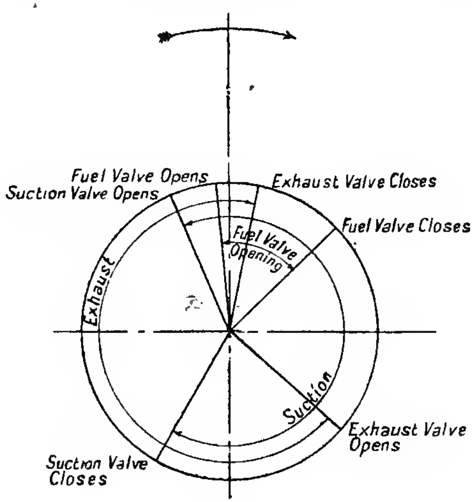


FIG. 4.

The Two Stroke Cycle.—As its name implies, the two stroke cycle is completed in one revolution of the engine. The revolution may roughly be divided into three nearly equal parts :—

(1) Combustion and Expansion.
(2) Exhaust and Scavenge.
(3) Compression.

The exhaust and scavenge take place when the piston is near the bottom dead centre, and consequently only very small portions of the expansion and compression strokes are lost, in spite of the fact that nearly 120° of the crank revolution are occupied with exhaust and scavenge. This point is clearly seen on reference to Fig. 5.

Exhaust Period.—The exhaust starts when the piston uncovers slots in the cylinder wall. The point at which this happens is different in different designs of engine, an average being about 15% of the stroke before bottom dead centre. The exhaust ports are usually of large area, and consequently the pressure falls to atmospheric very rapidly. The period required for this process naturally depends on the port area and the piston speed, and average figures are about 20° to 30°.

It is well to dwell carefully on the state of affairs at this point.

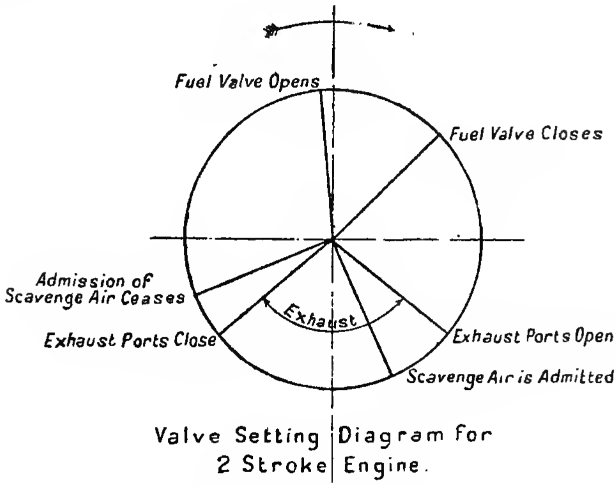


Fig. 5.

During exhaust the cylinder pressure has fallen from about 55 to about 15 lb. per sq. in. absolute, and there is no reason to suppose that the remaining exhaust gases have fallen greatly in temperature. (Given adiabatic expansion, the fall in absolute temperature is less than 20%.) The conclusions are, therefore :—

(1) Something like 50% by weight of the gases have effected their escape.

(2) The remaining gases are rarefied compared with atmospheric air.

Scavenge Period.—The scavenge air is admitted by ports or valves (or both), and the instant at which admission starts is timed to coincide with that at which the cylinder contents

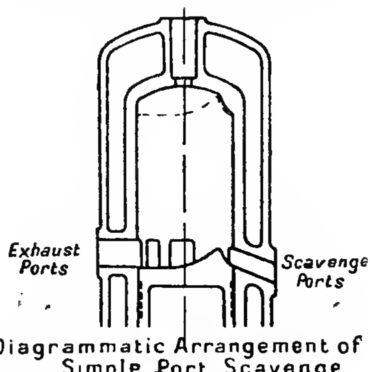
attain appreciably the same pressure as the scavenge air, or a trifle less. The incoming scavenge air is supposed to sweep the remaining exhaust gas before it and so fill the cylinder with a charge of pure air by the time the piston has covered the exhaust slots on the up stroke. Actually certain processes take place which do not enter into the ideal programme. Some of these are :—

(1) A certain amount of mixing between the incoming scavenge air and the retreating exhaust gases.
(2) Short circuiting of scavenge air to the exhaust pipe before all the exhaust gas has been expelled.

The effects of both these processes are minimised by providing a large excess of scavenge air. The figure adopted for the ratio of scavenger volume to cylinder volume was about 1·4 in the earlier designs, but later experience points to the advisability of increasing this figure to about 1·8.

There are a number of different systems in use for admitting scavenge air, and some of these are discussed below.

Simple Port Scavenge.—In this system the scavenge air is admitted by means of ports in the cylinder liner opposite a



Diagrammatic Arrangement of Simple Port Scavenge

FIG. 6.

row of similar ports for the exhaust (see Fig. 6), the piston top being provided with a projection to deflect the scavenge air to the top of the cylinder. This system is simple but possesses some disadvantages which are enumerated below.

(1) The scavenge air slots have to be made shorter than the exhaust slots in order that the cylinder pressure may fall to

the same value as the scavenge air pressure before the piston begins to uncover the scavenge slots. This entails the latter being covered by the piston on its upward stroke before the exhaust ports are covered, and consequently the pressure at the beginning of compression can barely exceed the pressure in the exhaust pipe. There is also a possibility of exhaust gases working back into the cylinder.

(2) The projection on the top of the piston necessitates an irregularly shaped cylinder cover in order to provide a suitable shape for the combustion space.

Engines provided with this system of scavenge are only suited for a relatively low mean indicated pressure of about 80 lb. per sq. in.

Cylinder Cover Valve Scavenge.—In this system the scavenge air is admitted by means of one to four valves located in the cylinder cover, and avoids some of the disadvantage of the simple port scavenge.

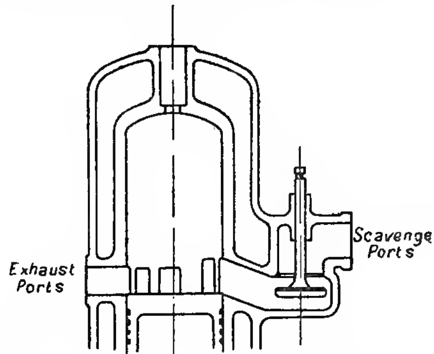
By allowing the scavenge valves to close after the exhaust ports have been covered by the piston, the cylinder may become filled with air at scavenge pressure before compression starts, and consequently such a cylinder is capable of developing a higher mean effective pressure. The greatest drawback to this system is the complication of the cylinder cover, and this appears to be rather serious. Strenuous efforts are being made to design suitable cylinder covers for this type of two stroke engine, but with the exception of one or two designs, which are now in the experimental stage, most of them appear to have a comparatively short life.

Valve Controlled Port Scavenge.—This system (usually associated with the name of Messrs. Sulzer Bros.) appears to combine most of the advantages of both with the disadvantages of neither of the above systems.

The air ports, or a certain number of the air ports, are so situated that they are uncovered before the exhaust ports, but are controlled by a valve of the double beat, piston, or other type, in such a way that communication does not exist between the cylinder and the scavenge pipe until the exhaust ports have been uncovered for a sufficient period to allow the cylinder pressure to fall to or below the scavenge pressure. On the upward stroke the controlling valve remains open, so that the cylinder is in communication with the scavenge pipe until the scavenge slots are covered by the piston. (See Fig. 7.)

Engines controlled on this principle are at present the most successful of the large two stroke Diesel Engines.

Amongst small or medium powered two stroke engines the simple port scavenge principle is the favourite.



Diagrammatic Arrangement of 2 Stroke Cylinder with Controlled Port Scavenge.

FIG. 7.

Types of Diesel Engines.—The existing types of Diesel Engines can be divided into groups in various ways, according as they are :—

(1) Stationary or Marine.
(2) Four Cycle or Two Cycle.
(3) Slow Speed or High Speed.
(4) Vertical or Horizontal.
(5) Single Acting or Double Acting.

It suffices here to describe shortly the outstanding features of the commonest types in commercial use.

Four Stroke Slow Speed Stationary Engines.—There are probably more Diesel Engines falling under this heading than any other. The usual arrangement is a vertical engine having one to six cylinders provided with trunk pistons. The main features are (1) a massive cast-iron bedplate made in one or two pieces, the bottom being formed in the shape of a tray to catch any lubricating oil which is thrown out of the bearings or drips from the cylinders. (2) Massive columns forming the cylinder jacket into which the liners are pressed. (3) A heavy fly-wheel of large diameter, necessitated by the low speed of revolution.

These are perhaps the most reliable Diesel Engines hitherto constructed, and differ but little from the machines made by the Maschinenfabrik Augsburg in the very earliest days of Diesel Engine manufacture. For economy in fuel oil, and lubricating oil consumption these engines are unsurpassed, and the massiveness of their construction secures for them a long, useful life.

Somewhat similar engines are made in horizontal form, in accordance with popular gas engine practice.

The horizontal engine is obviously suitable where head room is limited, and it is a trifle cheaper to manufacture than the vertical design. The latter, however, is still the general favourite, and is made in very large numbers in standard sizes, varying from 8 to 200 B.H.P. per cylinder.

Four Stroke High Speed Stationary Engines.—The high speed Diesel Engine was introduced in order to cheapen manufacture and to provide a more compact prime mover for use in limited spaces. Forced lubrication is usually adopted, and an enclosed crank-case is therefore provided. The fly-wheel is of course much smaller and lighter than that of a slow speed engine of the same power. In other respects the design follows fairly closely that of the slow speed engine only that additional care has to be exercised in the design of those details in which inertia plays an important part. The fuel consumption is practically the same as that of a slow speed engine, but the lubricating oil consumption is usually higher and the useful life of the engine somewhat shorter.

Two Stroke Slow Speed Stationary Engines.—These are built in sizes varying from about 200 to 700 B.H.P. per cylinder. Their chief advantage over the four cycle equivalent is reduced cost and size. The fuel consumption is slightly higher than that of the four stroke engines of the same power and the cost of upkeep would also appear to be slightly greater. These engines are usually fitted with a cross-head, and this enables forced lubrication to be used without an unduly high consumption of lubricating oil. The use of cross-heads is also becoming increasingly common with four cycle engines, and this appears to be a move in the direction of increased reliability.

Other Types of Stationary Engines.—Horizontal two stroke Diesel Engines have been made in small numbers. Examples of double-acting horizontal engines are also to be

found, but the three types described above appear to fulfil most requirements for land service.

Four Stroke Marine Engines.—Most of the successful marine Diesel Engines are of this type. Cross-head and guides are usually fitted and forced lubrication is preferred. The piston speed is usually moderate, being about 700 to 850 per minute. In Messrs. Burmeister, Wain's well-known design the bedplate and columns form an enclosed box construction of great rigidity and strength. The total enclosure of the crank-case is also conducive to economy of lubricating oil. In other respects the engines follow land practice very closely in almost every essential point of design. It has been frequently contended in the past that the failure of certain marine Diesel Engines has been almost entirely due to lack of knowledge of the requirements of marine service. The actual facts of the case appear to be that almost every complete failure of a marine Diesel Engine has been due to causes which would have led to the same result on land. Under the category of failures must be placed a number of large two stroke Diesel Engines, which have been installed on board ship before they had proved their reliability on land. The actual modifications required to convert a successful land design into a successful marine design are comparatively trivial. When once the essential difficulties of Diesel Engine construction have been successfully surmounted the adaptation to the requirements of marine service is a small matter in comparison.

Two Stroke Marine Engines.—As indicated in the previous paragraph, there have been many failures with large two stroke engines. In cases where the manufacturers have had no experience of marine work, they appear to have been unduly influenced in favour of accepted marine traditions. Where the engine has been constructed by marine steam engineers, under licence from a Diesel Engine manufacturing firm, the result has been similar. The machines resulting from this fusion of ideas have generally resembled marine steam engines with Diesel Engine cylinders. The columns, guides and bedplate are scarcely to be distinguished from those of a marine steam engine. The open crank-pit, adopted out of deference to steam engine practice, appears to lead to an excessive consumption of lubricating oil, and renders the economical use of forced lubrication impossible, besides facilitating the ingress of dirt. Another feature borrowed from

steam practice consists of the rocking levers driven off the cross-head for working the scavenge air pumps. These have not always been quite successful. These and other circumstances all seem to point to the advisability of developing the marine Diesel on lines a little more independent of steam engine precedent. At present the two stroke marine Diesel Engine, of large power, must still be considered to be in the experimental stage (without prejudice to two or three isolated cases of large two stroke Diesel Engines at present operating successfully). For moderate powers, up to about 500 H.P., two stroke engines have been fairly widely and successfully used.

High Speed Marine Engines.—A large number of designs of both four stroke and two stroke high speed marine Diesel Engines have been described in the technical press of recent years.

Many of these differ but little (except in the provision of a reversing gear and other details) from high speed land engines. Others, particularly of the two stroke type, have been developed along strongly individual lines for some special purpose, such as the propulsion of submarines.

To this latter class belong the stepped piston engines, in which an extension of the main piston serves at once as a cross-head guide and a single acting scavenge pump piston.

With one or two exceptions, these engines appear difficult to dismantle, and are not likely to become popular until this objection is removed.

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CHAPTER II

THERMAL EFFICIENCY

THE overall thermal efficiency of a heat engine is the ratio of the useful work performed to the mechanical equivalent of the heat supplied during a given period of working.

Problem : What is the thermal efficiency of a Diesel Engine which consumes 0·4 lb. of fuel per brake horse-power hour ?

To solve this problem two things require to be known :—

(1) How much heat 1 lb. of fuel gives out on combustion, i.e. the calorific value of the fuel.

(2) How much mechanical work is equivalent to a given quantity of heat.

The calorific value of different qualities of liquid fuel varies from about 15,300 (Mexican crude) to about 19,300 (Galician crude) British Thermal Units per lb. For calculations and comparison of test results fuel consumptions are usually reduced to their equivalents at a calorific value of 18,000 B.T.U. per lb. One B.T.U. (the amount of heat required to raise the temperature of 1 lb. of water 1° F.) is equivalent to 778 ft. lb. This is Joules' equivalent. Therefore, since 1 H.P. hour = 1,980,000 ft. lb., the required thermal efficiency is equal to :—

$$\frac{1,980,000}{0{\cdot}4 \times 18,000 \times 778} = 0{\cdot}35.$$

From this it is seen that roughly one-third of the heat supplied has been converted into useful work. It is within the province of thermodynamics to determine what proportion of the heat loss is theoretically unavoidable, and to what extent the performance of an actual engine approximates to that of an ideal engine working on the same cycle of operations. It is not proposed to give here more than a brief summary of the physical laws relating to the behaviour of air under the in-

fluence of pressure and temperature, which form the basis of thermodynamic investigations.

Pressure, Volume and Temperature of Air.—The relation between these three quantities is expressed by the formula :—

$P.V. = 53{\cdot}2 \times w \times T$ —(1) where P=Pressure in lb. per sq. ft. abs.
V=Volume in cubic ft.
w=Weight in lb. of the quantity of air under consideration.
T=Temperature in degrees abs. F.

This relation holds good for any condition of temperature and pressure, and for a specified weight of air, given the values of any two of the quantities represented by capital letters, the third can be calculated.

Example : Find the volume of 1 lb. of air at atmospheric pressure and 60° F. In this case $P=14{\cdot}7$ lb. per sq. in. abs., $T=60+461=521°$ abs. F., and $w=1$.

$$\text{Hence : } V=\frac{53{\cdot}2 \times 521}{14{\cdot}7 \times 144}=13{\cdot}1 \text{ cub. ft.}$$

Isothermal Expansion and Compression.—If the temperature remains constant during compression or expansion the process is said to be isothermal, and the value of T in equation (1) becomes a constant quantity. Thence for isothermal processes equation (1) becomes : $P.V.=\text{constant}$———(2).

If 1 lb. of air is under consideration the value of the constant is equal to $53{\cdot}2$ times the absolute temperature.

Work done during Isothermal Compression.—If P_1 and V_1 represent the pressure and volume before compression, and P_2 and V_2 the same quantities after expansion, then :—

$$\text{Work done}=\text{const. } \log_e \frac{V_1}{V_2} \text{———(3),}$$

the constant being that of equation (2). It should be borne in mind (though the bare fact can only be stated here) that the internal energy of a gas depends on its temperature only, regardless of the pressure. It therefore follows that all the work done in isothermal compression must pass away as heat through the walls of the containing vessel, and for this reason

isothermal processes are not attainable in practice, though they may be approximated to by slow compression in cylinders arranged for rapid conduction of heat.

Specific Heat at Constant Volume.—If 1 lb. of air is heated in a confined space, i.e. at constant volume, 0·169 B.T.U. are required to raise the temperature by 1° F. This then is the specific heat of air at constant volume. For many purposes it is near enough to consider the specific heat as constant, though actually its value increases slightly as the temperature increases. The amount by which the internal energy of 1 lb. of air increases as the temperature rises is therefore :—

$0{\cdot}169\,(T_2-T_1)$ where (T_2-T_1) = the increase of temperature.

Specific Heat at Constant Pressure.—In this case, on the other hand, work is done by expansion if heat is being added, and by compression if heat is being discharged ; consequently the specific heat at constant pressure exceeds that at constant volume by the equivalent of the work done. The specific heat at constant pressure is 0·238 B.T.U. per lb. per degree Fahrenheit.

Adiabatic Expansion and Compression.—Expansion or compression unaccompanied by the transfer of heat to or from the air is termed Adiabatic. It should be noted that the air may lose heat by doing external work or gain heat by having work done on it by the application of external force during an adiabatic process ; but this heat comes into existence or passes out of existence within the air itself and does not pass through the walls of the containing vessel.

The following relation holds good between the pressure and the volume during an adiabatic process :—

$P\,V^n = \text{constant}$ ——(4) where n = ratio of the two specific heats, viz. —1·41.

Owing to the conductivity of the cylinder walls adiabatic compression and expansion are not to be obtained in practice, but it is frequently possible to express the actual relation between pressure and volume by means of equation (4) if suitable values are chosen for "n." A little consideration will shew that for compression with some loss of heat the value of "n" will be less than 1·41, and for expansion with some loss

of heat greater than 1·41. An expansion or compression in which the relation between P and V is expressed by equation (4) is known as a Polytropic Process.

Work done on Polytropic Expansion of Air.—The work done is the integral of the pressure with respect to the volume, and if expressed in ft. lb. is given by :—

$$W = \frac{P_1V_1 - P_2V_2}{n-1} = 53 \cdot 2 \; \frac{(T_1 - T_2)}{n-1} \text{———} (5)$$

for 1 lb. of air, where the suffixes 1 and 2 have their usual significance in denoting the state before and after expansion ; the work done during a corresponding compression between the same pressures is of course the same.

Temperature Change during Polytropic Expansion.—By combining equations (1) and (2) the following is obtained :—

$$\frac{T_2}{T_1} = \left(\frac{P_2}{P_1}\right)^{1-\frac{1}{n}} = \left(\frac{V_1}{V_2}\right)^{n-1} \text{———} (6)$$

With the information supplied by the above six equations it is possible to construct the Indicator Diagram corresponding to an Ideal Diesel Engine, in which all defects of combustion and heat losses to the cylinder walls are supposed to be eliminated.

An Ideal Diesel Engine.—Before proceeding further, it is necessary to define what is to be understood by the term ideal engine for the purposes of this investigation. In the first place, frictional losses and all leakage are eliminated and compression and expansion are supposed to take place adiabatically ($n = 1 \cdot 41$). The specific heats are supposed to be constant and the same whether for air or a mixture of air or exhaust gases. On the other hand, the provision of compressed air for the purpose of fuel injection will be recognised, and consequently the machine will have a mechanical efficiency less than unity.

The compression in the air compressor will be regarded as isothermal and the compressor itself free of all mechanical or other losses.

It will also be supposed that all the exhaust gas, including that contained in the clearance space, is ejected on the exhaust stroke and that every suction stroke fills the cylinder with air at 14·7 lb. per sq. in. abs. and 521° abs. F. The stroke volume is taken for convenience of calculation at 100 cub. ft.

The Ideal Indicator Card.—The indicator card corresponding to this ideal engine is now readily constructed, and is shown in Fig. 8.

Point A denotes a volume of air equal to 100 cub. ft. plus the clearance space, which has to be calculated.

Line AB represents adiabatic compression from 14·7 lb. per sq. in. to 514·7 per sq. in. abs.

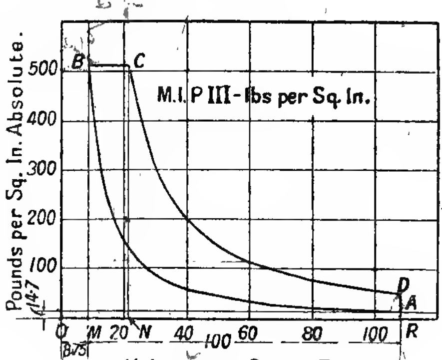


FIG. 8.

Line BC represents increases of volume at constant pressure due to addition of heat by combustion of fuel.

Line CD represents adiabatic expansion of the products of combustion and DA the fall in pressure due to exhaust release. It is not necessary to deal with the exhaust and suction strokes, as these are supposed to take place at atmospheric pressure.

Determination of Clearance Value.—

From equation (4).

$$\left(\frac{V_A}{V_B}\right)^{1.41} = \frac{P_B}{P_A} = \frac{514\cdot7}{14\cdot7} = 35$$

$$\therefore \quad \frac{V_A}{V_B} = \left(\frac{1}{35}\right)^{\frac{1}{1\cdot41}} = 12\cdot45$$

$$\therefore \quad \frac{100}{V_B} = 11\cdot45 \quad \therefore \quad V_B = 8\cdot75 \text{ cub. ft.}$$

This determines Points B and A.

Construction of Line AB.—This is effected by calculating the pressure at various points of the stroke in accordance with the following schedule :—

1	2	3	4	5	6	7
Per cent of stroke.	V= cubic feet.	$\frac{V_a}{V}$	Log $\frac{V_a}{V}$	(4)×1.41.	Antilog (5)	P=(6)×14·7 lb. per sq. in.
0	108·75	1·00	0·000	0·000	1·00	14·7
20	88·75	1·23	0·099	0·127	1·34	19·7
40	68·75	1·58	0·199	0·280	1·90	27·9
60	48·75	2·23	0·348	0·490	3·09	45·5
80	28·75	3·78	0·579	0·815	6·53	95·5
90	18·75	5·79	0·763	1·075	11·89	174·5
95	13·75	7·90	0·898	1·265	18·41	270·6
100	8·75	12·45	1·095	1·545	35·08	514·7

This simple calculation is given in detail, as it is typical.

Determination of Point c.—The value of V_c depends of course on the amount of heat added. The effect of the oil itself in increasing the weight of the working fluid will be ignored.

The blast air, however, will be taken into consideration and assumed to be equivalent to 8 cub. ft. of free air.

The weight of air concerned has now to be calculated.

(1) Suction air.

From equation (1) $w=\frac{P\times V}{53\cdot2\times T}=144\times\frac{14\cdot7\times108\cdot75}{53\cdot2\times521}=8\cdot3\text{ lb.}$

(2) Blast air.

$$\text{Weight of blast air}=\frac{8\cdot3\times8}{108\cdot75}=0\cdot61\text{ lb.}$$

„ „ suction air + blast air $=8\cdot3+0\cdot61=8\cdot91$ lb.

The effect of the heat released by the combustion of the fuel is to increase the temperature of :—

(1) The suction air (8·3 lb.),
(2) The blast air (0·61 lb.),

at a constant pressure of 514·7 lb. per sq. in. abs.

The temperature of the blast air is taken to be 521° F. abs., and that of the adiabatically compressed air is found from equation (6) as follows :—

$$\frac{T_B}{T_A}=\left(\frac{P_B}{P_A}\right)^{1-\frac{1}{n}}=\left(\frac{514\cdot7}{14\cdot7}\right)^{\cdot29}=2\cdot805$$

Therefore $T_B=521\times2\cdot805=1460°$ F. abs.

Now let H = heat added in B.T.U.
Then since the pressure remains constant,

$$H = \cdot 238\ [8\cdot 3\ (T_c - 1460) + \cdot 61\ (T_c - 521)]$$
$$= 2\cdot 12\ T_c - 2961$$

from which $$T_c = \frac{H + 2961}{2\cdot 12} \text{——} (7)$$

and from equation (1)

$$V_c = \frac{T_c \times 53\cdot 2 \times 8\cdot 91}{514\cdot 7 \times 144} = \cdot 0064\ T_c \text{——} (8)$$

Now suppose that 0·2 lb. of fuel is added, having a calorific value of 18,000 B.T.U. per lb., the resulting temperature will be :—

$$\frac{0\cdot 2 \times 18{,}000 + 2961}{2\cdot 12} = 3{,}100°\ \text{F. abs.}$$

and $V_c = \cdot 0064 \times 3100 = 19\cdot 8$ cub. ft.

The expansion line CD can now be constructed in the same way as the compression line, both being adiabatics. The value of the terminal pressure P_d is required for calculating the work done, and is found by means of equation (4) as follows :—

$$\frac{P_d}{P_c} = \left(\frac{V_c}{V_d}\right)^{1\cdot 41} = \left(\frac{19\cdot 8}{108\cdot 75}\right)^{1\cdot 41} = \cdot 091$$

$$\therefore\ P_d = \cdot 091 \times 514\cdot 7 = 46\cdot 8 \text{ lb. per sq. in. abs.}$$

Calculation of Work Done.—Referring to Fig. 8, the work done is clearly equal to the areas BCNM plus CDRN less BARM. Using equation (5) for the two latter, we have :—

Area BCNM $= 144 \times 514\cdot 7 \times (19\cdot 8 - 8\cdot 75) = 819{,}000$ ft. lb.

,, $$\text{CDRN} = \frac{144\ (514\cdot 7 \times 19\cdot 8 - 46\cdot 8 \times 108\cdot 74)}{\cdot 41} = 1{,}800{,}000 \text{ ft. lb.}$$

By addition 2,619,000 ,, ,,

,, $$\text{BARM} = \frac{144\ (514\cdot 7 \times 8\cdot 75 - 14\cdot 7 \times 108\cdot 75)}{\cdot 41} = 1{,}020{,}000 \text{ ,, ,,}$$

Work done by difference = 1,599,000 ,, ,,

Since 1 H.P. hour = 1,980,000 ft. lb.
the fuel consumption is :—

$$\frac{0\cdot 2 \times 1{,}980{,}000}{1{,}599{,}000} = \cdot 248 \text{ lb. per I.H.P. hour.}$$

From equation (3) the work done in compressing the blast air is given by

$$8 \times 14{\cdot}7 \times 144 \times \log_e \frac{914{\cdot}7}{14{\cdot}7} \text{ (assuming blast pressure of 900 lb. per sq. in.).}$$

$$= 70{,}000 \text{ ft. lb.}$$

so the nett work done is

$$1{,}599{,}000 - 70{,}000 = 1{,}529{,}000 \text{ ft. lb.}$$

and the consumption of fuel per B.H.P. hour is :—

$$\frac{0{\cdot}2 \times 1{,}980{,}000}{1{,}529{,}000} = {\cdot}259 \text{ lb. per B.H.P. hour.}$$

The mechanical efficiency being $\frac{1{,}529{,}000}{1{,}599{,}000} = 95{\cdot}6\%$

The M.I.P. of the diagram is equal to the indicated work divided by the stroke volume.

$$\frac{1{,}599{,}000}{100} = 15{,}990 \text{ lb. per sq. ft.}$$

$$= 111 \text{ lb. per sq. in.}$$

This is a trifle higher than the M.I.P. usually indicated at full load in an actual commercial engine.

The brake thermal efficiency is given by :—

$$\frac{1{,}980{,}000}{0{\cdot}259 \times 18{,}000 \times 778} = 0{\cdot}548$$

A splendid figure for the brake thermal efficiency of an actual engine of large size is 0·35 ; comparing this with the above figure, it is seen that the actual engine attains 64% of the efficiency attributed to the ideal. The indicated thermal efficiency of the ideal engine is :—

$$\frac{1{,}980{,}000}{0{\cdot}247 \times 18{,}000 \times 778} = 0{\cdot}573$$

That of an actual engine at full power is about 0·472, the ratio of actual to ideal being about 82%. From the above it will be seen that so far as the thermal actions within the cylinder are concerned, an actual Diesel Engine in good order leaves comparatively little room for improvement as long as the accepted cycle is adhered to. As a matter of fact, slight deviations from the constant pressure cycle are frequently made, and improved efficiencies obtained thereby. Most high speed Diesels, for example, are arranged to give an indicator

card, showing a certain amount of explosive effect, i.e. combustion at constant volume, causing the pressure at the dead centre to rise to a figure which may be anything up to about 100 lb. above the compression pressure. This is found to have a beneficial effect on the fuel consumption, which is readily explained on theoretical grounds; but it is obvious that considerations of strength must limit the extent to which this principle is used.

Fuel Consumption at Various Loads.—If the foregoing calculations for the fuel consumption of the ideal engine are

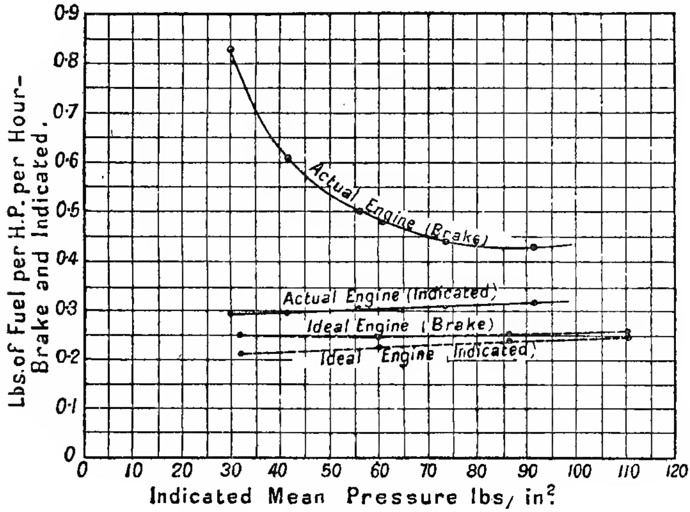


FIG. 9.

repeated for various values of the quantity of fuel admitted results will be obtained which are shown graphically in Fig. 9, together with test results of an actual engine. The fuel consumptions per I.H.P. and B.H.P. hour are plotted on a basis of M.I.P.; the actual engine to which the test results refer was of the four stroke type developing 100 B.H.P. per cylinder at full load. There are two facts to be noticed :—

(1) The fuel consumption per I.H.P. hour increases as the M.I.P. increases.

(2) The fuel consumption per B.H.P. hour attains a minimum value at about 90 lb. M.I.P. in the case of the actual engine

and about 60 lb. in the case of the ideal. The reason for (1) will be evident on comparing theoretical diagrams for various values of the M.I.P., and the case is quite comparable to that of a steam engine working with an early cut-off.

The point of maximum brake efficiency depends upon two conflicting influences, viz. the indicated efficiency which decreases and the mechanical efficiency which increases as the M.I.P. is augmented.

It is usual in this country, when considering mechanical efficiency, to treat the work done in driving the air compressor as a mechanical loss, so that :—

$$\text{Mechanical efficiency} = \frac{\text{B.H.P.}}{\text{I.H.P. of main cylinders.}}$$

Continental engineers, on the other hand, sometimes subtract the indicated power of the compressor from that of the main cylinders for the purposes of the above equation.

The usual British practice will be adhered to throughout this book.

Variation of Mechanical Efficiency with Load.—Examination of a large number of Diesel Engine test results reveals the fact that the difference between the I.H.P. and the B.H.P. remains nearly constant as the load is varied. This fact enables one to calculate the mechanical efficiency at any load if the full load efficiency is known.

Example: What is the mechanical efficiency at three-quarter, half and quarter load if that at full load is 72% ? Let the full load I.H.P. be represented by 100, then the following tabulated figures hold good :—

	B.H.P.	I.H.P.	Const. Diff.	Mech. Efficy.
Full load	72	100	28	·72
Three-quarter load .	54	82	28	·66
Half load	36	64	28	·56
Quarter load . . .	18	46	28	·39

This method yields sufficiently accurate results for most estimating purposes.

Mechanical Losses.—The work corresponding to the difference between the I.H.P. and the B.H.P. is approximately accounted for in the two following tables, which apply to

medium-sized engines of good design, and of the four stroke and two stroke types respectively :—

FOUR STROKE ENGINE—Full Load Mech. Efficy., 75%.

	per cent
Brake-work	75·0
Work done on suction and exhaust strokes	3·0
Indicated compressor work	5·8
Compressor friction	1·2
Engine friction, opening valves, etc.	15·0
Work indicated in main cylinders	100·0

TWO STROKE ENGINE—Full Load Mech. Efficy., 70%.

	per cent
Brake-work	70·0
Indicated compressor work	6·5
Compressor friction	1·4
Indicated scavenger work	7·5
Scavenger friction	1·6
Engine friction, opening valves, etc.	13·0
Work indicated in main cylinders	100·0

Improvements in bearings and guides on the principle of the well-known Michell bearing or the use of roller bearings for the main journals and big ends suggest possibilities for reducing engine friction which will possibly materialise in the future. The adoption of some form of limit piston ring to prevent excessive pressure on the liners would also help matters in the same direction, besides increasing the life of the liners.

Influence of Size on Mechanical Efficiency and Fuel Consumption.—Piston speed, within the range of present practice, appears to have very little influence on either mechanical efficiency or fuel consumption. This remark does not apply to the abnormally low speeds obtaining, for example, with a marine engine turning at reduced speed.

The cylinder bore is the principal factor in economy, always assuming a reasonable ratio of bore to stroke and good design generally.

The following table is a rough guide to the mechanical efficiency and fuel consumptions to be expected from cylinders of various sizes working at full load.

Four Stroke				Two Stroke			
Cylinder Diam., in.	Mech. Effcy. %	Fuel per B.H.P. hr. lb.	Fuel per I.H.P. hr. lb.	Cylinder Diam., in.	Mech. Effcy. %	Fuel per B.H.P. hr. lb.	Fuel per I.H.P. hr. lb.
10	·70	·46	·320	10	·65	·51	·332
15	·73	·43	·315	15	·68	·48	·326
20	·75	·41	·308	20	·70	·46	·322
25	·76	·40	·305	25	·71	·45	·320
30	·76	·40	·305	30	·71	·45	·320

The fuel consumption per B.H.P. at loads other than full load is readily found by first calculating the probable mechanical efficiency, as described in the previous article, and then allowing for a fall in the consumption per I.H.P. proportional to that shown on Fig. 9 for a typical engine for the same M.I.P.

Heat Balance Sheet.—An elaborate trial of a Diesel Engine includes the measurement of the quantity of cooling water used, the inlet and outlet temperatures of the water and the temperature of the exhaust gases. Apart from very slight losses, such as radiation, etc., these data usually enable the heat supplies by the fuel to be accounted for.

A typical heat balance is given below :—

Accounted for by indicated work	42%
Rejected to cooling water	34%
Rejected to exhaust	24%
Total heat supplied . . .	100

A striking feature of this balance is the large amount of heat appearing on the cooling water account, which at first sight would appear to indicate very poor utilisation of heat within the cylinder. It has been shown that so far from this being the case, an actual engine in good order indicates about 80% of the work attributable to an ideal engine.

The explanation of this lies in the fact that a large proportion of the heat received by the cooling water is given out by the exhaust gases after combustion is complete, particularly on their passage through the cylinder cover in the case of a four stroke engine and through the ports in the case of a two cycle. Most of the friction work done by the piston and all the compressor work appear on the cooling water account.

Efficiency of Combustion.—In all actual oil engines there is a considerable amount of "after burning," i.e. gradual burning during the expansion stroke. In a well-tuned Diesel Engine this effect is not sufficiently pronounced to cause smoke even at considerable overload, 120 lb. per sq. in. M.I.P. for example. Exaggerated "after burning" is to be avoided as, in addition to increasing the fuel consumption, it increases the mean temperature of the cycle and of the exhaust stroke particularly, and gives rise to accentuation of all the troubles which arise from the effects of high temperature. The most prominent of these troubles are enumerated below :—

(1) Cracking of piston crown and cylinder covers.
(2) Pitting of exhaust valves.
(3) Increased difficulty of lubricating the cylinders, resulting in—
  (a) Increased liner wear.
  (b) Sticking of piston rings.
  (c) Liability to seizure of piston.
(4) Increase of temperature of gudgeon pin bearing.

The above formidable list is probably not exhaustive, but is sufficient to show the desirability of securing the best possible conditions, apart altogether from the question of economy in fuel and lubricating oil consumption. The attainment of good combustion, assuming a good volumetric efficiency of cylinder and good compression, depends more than anything upon small points in connection with the fuel valve, which are easily adjusted on the test bed, provided the design of the fuel valve is satisfactory.

Entropy Diagrams.—Entropy is a convenient mathematical concept which it is difficult, and perhaps impossible, to define in non-mathematical terms. It is sometimes described as that function of the state of the working fluid which remains constant during an adiabatic process. Entropy increases when heat is taken in by the working fluid and decreases when it is rejected. If heat is supplied to the working fluid at constant temperature, then the increase of entropy is equal to the amount of heat so supplied divided by the absolute temperature. If the temperature is variable during the process of heat absorption, then the increase of entropy is determined by the integration of the equation :—

$$d\phi = \frac{dH}{T}$$ where ϕ = Entropy.
H = Heat taken in or given out.
T = Absolute temperature.

The zero of entropy may be located at any convenient level of temperature except absolute zero. It is convenient for our purposes to consider the entropy to be zero when

$$P = 14{\cdot}7 \text{ lb. per sq. in. abs.}$$
$$T = 521^\circ \text{ F.}$$

The value of a diagram connecting T and ϕ during the working cycle of an internal combustion engine depends upon the following properties :—

(1) Increasing and diminishing values of ϕ denote heat supplied and heat rejected respectively.

(2) For a complete cycle the diagram is a closed figure, the area of which is proportional to the quantity of heat which has been converted into work during the cycle.

The area of the diagram is given by :—

$$\int T\, d\phi = \int \frac{TdH}{T} = H_1 - H_2$$ the difference between the heat supplied and that rejected.

Construction of an Entropy Chart for Air (see Fig. 10).—The axis of T is drawn vertically, and includes temperatures from 0 to 4000° F. abs. The axis of ϕ is horizontal, and is graduated from 0 to 0·35. It will be clear that the axis of T is an adiabatic line of zero entropy and that any selected temperature on this line corresponds to a definite pressure, which can be calculated by means of equation (6). Actually it is more convenient to tabulate a series of pressures from 0 to, say, 700 lb. per sq. in. abs., and tabulate the corresponding temperatures. Points on the axis of T found in this manner are the starting points of constant pressure lines.

Constant Pressure Line. P=14·7 lb. per sq. in.—A constant pressure line is a curve connecting T and ϕ when P remains constant. It is usual to consider 1 lb. of air, and if the specific heat at constant pressure is assumed to be 0·25

Then $dH = 0{\cdot}25\ dT$

$$\text{And } d\phi = \frac{dH}{T} = {\cdot}25\ \frac{dT}{T}$$

$$\text{Therefore } \phi = {\cdot}25 \log_e \frac{T_2}{T_1} \text{—} T_1 \text{ being } 521^\circ \text{ F. abs.}$$

Selecting various increasing values of T_2 corresponding values of ϕ are calculated and plotted on the chart, a fair curve passing through the points being the required constant pressure line. One such line having been constructed, lines corresponding to other pressures are readily drawn, since their ordinates

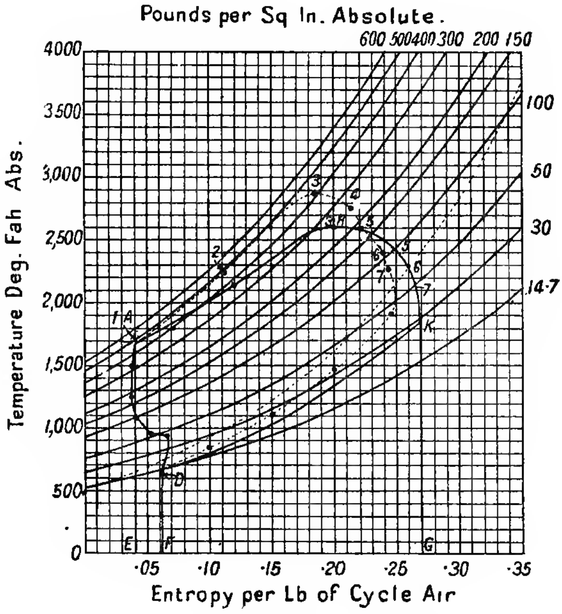


FIG. 10.

are proportional to the temperatures indicated by their starting points.

Constant Volume Lines.—These can be constructed in a similar manner, using the specific heat at constant volume instead of that of constant pressure. On Fig. 10 only one constant volume line is shown, viz. that corresponding to

$$P=14{\cdot}7 \text{ lb. per sq. in.}$$
$$T=673^\circ \text{ F. abs.}$$

as this is required to complete the diagram by representing the rejection of heat at constant volume at the end of the stroke. This process is of course a scientific fiction, as the pressure in an actual cylinder is reduced to approximately atmospheric pressure by the discharge of exhaust gases and not by cooling

the latter. The reason for using the value 0·25 for the specific heat must now be explained. Although the specific heat of pure dry air is 0·238, that of the gases present in the cylinder of a Diesel Engine is a variable quantity for the following reasons :—

(1) The composition of the working fluid is altered by the addition of the fuel.
(2) The specific heat of the exhaust gases increases slightly with increase of temperature.

If these variations were rigidly taken into account the construction of the entropy diagram would be a very laborious business, and the work is greatly curtailed by adopting certain approximations which will be described. In the first place the specific heat is assumed to be constant and equal to the calculated specific heat of the exhaust gases at 60° F. The variation in the weight of the working fluid is dealt with by first treating the diagrams as though the weight were constant and then correcting the diagram (entropy) by increasing the entropy and decreasing the absolute temperature of points on the expansion line in the same proportion by which the fuel and blast air increase the weight of the charge.

Use of the Entropy Chart.—A method of constructing the entropy diagram, corresponding to an indicator card, will now be described. The clearance volume of the engine must first be ascertained and the card accurately calibrated. Points on the indicator diagram are then marked, corresponding to every 15 or 30° or other convenient division of revolution of the crank past the top dead centre. Each of the 12 or more points so marked is given a reference number and the absolute pressure in lb. per sq. in., and also the volume in any arbitrary units (hundredths of an inch on the diagram, for instance) corresponding to each point is read off and tabulated. The apparent temperature corresponding to each point is then calculated by means of equation (1) on the assumption that the initial temperature just before compression is, say, 673° F. abs.

Points on the entropy diagram corresponding to the selected points on the indicator diagram are now found by following the appropriate constant pressure lines until the calculated temperatures are reached, thus obtaining the apparent entropy diagram, which requires to be corrected in accordance with the preceding article.

The entropy diagram shown in Fig. 10 has been constructed in the manner described, and the data are given below :—

M.I.P. shown by indicator diagram, 82 lb. per sq. in. (see Fig. 11).

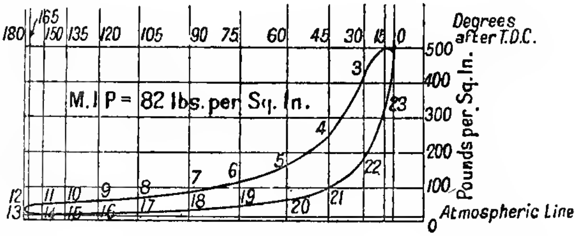


FIG. 11.

Initial temperature of suction air before compression assumed to be 673° F. abs.

Absolute temperature (apparent) calculated from PV=kT where P=Pressure in lb. per sq. in. abs. (scaled off diagram)

V=Volume measured in linear inches off diagram

T=Temperature in degrees F. abs.

Initial conditions are P=14·7, V=6·38, T=673.

Whence k=·1395.

Figures are given in the table below :—

Ref. No.	Degree after firing centre.	V. in in.	P. in lb. per sq. in.	Apparent Temperature.
1	0	0·48	490	1690
2	15	0·63	498	2260
3	30	0·95	421	2880
4	45	1·53	251	2760
5	60	2·17	163	2540
6	75	2·94	114	2400
7	90	3·75	84	2260
8	105	4·56	65	2130
13	180	6·38	14·7	673
19	265	2·94	44	930
20	300	2·17	61	950
21	315	1·53	99	1090
22	330	0·95	184	1250
23	345	0·63	331	1500

The apparent entropy diagram was plotted from the above values of P and T, and the corrected diagram, constructed in accordance with the preceding article, is shown in Fig. 10.

Area of corrected entropy diagram 9·90 sq. in.

Temperature scale . . . 1 in. = 500°.
Entropy scale . . . . 1 in. = ·05°.

Therefore work done per lb. of suction air is given by :—

$$9{\cdot}90 \times 500 \times 0{\cdot}05 = 248 \text{ B.T.U.}$$

1 lb. of suction air @ 673° F. abs. occupies 16·9 cub. ft. Since clearance volume = 8% of stroke volume, therefore corresponding stroke volume = 16·9 ÷ 1·08 = 15·65 cub. ft. Therefore work done by suction air, according to the indicator card, @ 82 lb. per sq. in. M.I.P. is given by :—

$$\frac{82 \times 144 \times 15{\cdot}65}{778} = 238 \text{ B.T.U.}$$

This figure is about 4% less than that shown by the entropy chart, and suggests that perhaps the assumed temperature of the suction air before compression (viz. 673° F. abs.) is too high.

Note.—The low pressures at the beginning of the compression stroke and the end of the expansion stroke have not been used in the construction of the entropy chart for the following reasons :—

(1) Low pressures are very difficult to scale off the indicator card with any degree of accuracy.

(2) The low pressures indicated on the card are invariably erroneous, unless the greatest care has been exercised in taking the card, and that with a suitable indicator in perfect order. Many of the reputable makes of indicator, though quite suitable for steam engine work, give very inaccurate results when applied to internal combustion engines.

The following points should be noticed :—

(1) The compression line deviates from the true adiabatic in a manner which indicates that heat has been lost to the cylinder walls.

(2) The expansion does not become adiabatic until well after half stroke, showing that there is a certain amount of "after burning."

Area of Entropy Diagram.—If the diagram has been carefully done the area of the diagram in heat units should correspond fairly closely to the work shown on the indicator diagram. Deviations from equality may be due to :—

(1) Variation in the specific heat at high temperatures.
(2) Incorrect assumption of the initial temperature of the induced air.

Owing to the approximations referred to above there is generally a discrepancy of a small percentage between the amounts of work shown by the indicator card and the entropy diagram. Also the total area under the upper boundary of the entropy diagram should correspond to the total heat supplied per lb. of suction air less the heat discarded to the water-jacket on the expansion stroke. Investigation of the sort described seems to indicate that not more than about 10% to 15% of the total heat supplied by the fuel is lost in this way at 100 lb. M.I.P.

Specific Heat of Exhaust Gases.—The specific heat of a mixture of gases is the sum of the products of the specific heats of the constituents into the proportion by weight in which the constituents are present.

The specific heats (at constant pressure) of the constituent gases present in the exhaust of an oil engine are given below :—

Water vapour . . .	0·480	(Varies considerably with temperature)
Nitrogen	0·247	
Oxygen	0·217	
CO_2	0·210	(Varies considerably with temperature)
CO	0·240	

The average composition of air and the specific heat derived from that of its constituents are given below :—

N_2 . .	75·7% $\times$ 0·247	= ·187
O_2 . .	22·7 $\times$ 0·217	= ·049
H_2O . .	1·5 $\times$ 0·480	= ·007
Total .	100	·243

This is about 3% higher than the accepted value for pure dry air.

The approximate composition of exhaust gases assuming complete combustion is readily calculated as follows :—

Data M.I.P. – 100 lb. per sq. in.

Fuel per I.H.P. hour—0·31 lb.

Composition of fuel (assumed). . $\left\{\begin{array}{l}\text{Carbon, } 86\% \\ \text{Hydrogen, } 13\% \\ \text{Oxygen, } 1\%\end{array}\right\}$ by weight.

One H.P. hour = 1,980,000 ft. lb.

$\therefore$ Volume swept by piston per I.H.P. hour $= \dfrac{1{,}980{,}000}{100 \times 144}$

$= 137{\cdot}5$ cub. ft.

Clearance volume, say, 8% = 11·0 ,, ,,

Total weight of suction air = 148·5 (assuming perfect scavenge of clearance space).

Weight of suction air $= \dfrac{148{\cdot}5 \times 14{\cdot}7 \times 144}{53{\cdot}2 \times 673} = 8{\cdot}78$ lb.

@ 212° F.

Free volume of blast air @ 8% of stroke volume @ 60° F.
$= {\cdot}08 \times 137{\cdot}5 = 11$ cub.

Weight of blast air $= \dfrac{11 \times 147 \times 144}{53{\cdot}2 \times 521} = 0{\cdot}83$ lb.

$\therefore$ Total air	$= 8{\cdot}78 + 0{\cdot}83 = 9{\cdot}61$ lb.	= 97%
Weight of fuel .	= 0·31 ,,	= 3%
Total mixture .	9·92 ,,	100%

Composition of mixture before combustion is given by :—

Air	N_2 . . .	$\cdot 76 \times 97 = 73{\cdot}7\%$
	O_2	$(22{\cdot}7 \times 97) + (\cdot 01 \times 3) = 22{\cdot}0\%$
	H_2O . .	$\cdot 015 \times 97 = 1{\cdot}5\%$
Fuel	C . . .	$\cdot 86 \times 3 = 2{\cdot}5\%$
	H_2 . . .	$\cdot 13 \times 3 = 0{\cdot}4\%$
		100·1%

Combustion of carbon and hydrogen takes place in the proportions given by :—

$$C + O_2 = CO_2$$
$$12 + 32 = 44 \text{ by weight}$$

And

$$H_2 + O = H_2O$$
$$2 + 16 = 18 \text{ by weight}$$

The following table shows the composition after combustion and the specific heat (constant pressure) of the mixture :—

	Per cent.		Specific heat.		Product.
N_2	73·7	×	·247	=	·1820
CO_2 $2·5 \times 44 \div 12$ . .	9·2	×	·217	=	·0191
O_2 $22-(32 \times 2·5 \div 12)$ $-(16 \times 0·4 \div 2)$.	12·1	×	·210	=	·0254
H_2O $1·5-(18 \times 0·4-2)$.	5·1	×	·480	=	·0245
	100·1				

Specific heat of exhaust gases = ·2510

Repeating the above calculation for different values of the M.I.P., the following figures are obtained :—

M.I.P. lbs. per sq. in.	0	30	45	75	105	130	155
Specific heat . .	·245	·247	·248	·250	·252	·254	·256

Literature.—For thermodynamics of the internal combustion engine consult :—

Wimperis, H. E., "The Internal Combustion Engine."

Judge, A. W., "High Speed Internal Combustion Engines."

Clerk, Sir Dugald, "The Thermodynamics of Gas, Petrol, and Oil Engines."

For test results see :—

Chalkley, A. P., "The Diesel Engine."

Dalby, W. E., "Trials of a Small Diesel Engine."—Inst. N. A., April 2nd, 1914.

Wilkins, F. T., "Diesel Engine Trials."—Inst. M. E., October 20th, 1916.

— "Description and Tests of 600 B.H.P. 2 stroke cycle direct reversible Diesel Marine Engine."—*Engineering*, Dec. 22nd, 1916.

CHAPTER III

EXHAUST, SUCTION AND SCAVENGE

Renewal of the Charge.—In the theoretical study of heat engines the charge of working fluid is supposed to remain enclosed in a working cylinder and to undergo a cycle of physical changes due to the introduction of heat from, and discharge of heat to, external sources by conduction through the walls. In actual engines of the internal combustion type one constituent of the charge, viz. the oxygen, takes an active part in the chemical processes which constitute the source of energy. In such engines, therefore, the air charge must be renewed periodically. In all existing types of oil engine the charge is renewed as completely as possible for each cycle of thermal changes.

The way in which this is done in four stroke and two stroke Diesel Engines has been described in general terms in Chapter I. In the present chapter it is proposed to discuss the questions of the discharge of exhaust gases and the introduction of a new air charge from the quantitative point of view, apart from the consideration of the mechanical details, such as valves, cams, etc., of which these processes involve the use.

At the outset it will be necessary to state in a form convenient for application, the laws governing the flow of gases through orifices.

Flow of Gases through Orifices.—Fig. 12 is intended to represent a chamber containing gas at a pressure and temperature maintained constant at the values P_1 and T_1, and from which a gaseous stream is issuing through an orifice into the surrounding space, which is filled with gas at a constant pressure P_2.

In the first instance, it will be supposed that P_1 is very much greater than P_2. Then, according to the elementary theory of the flow of gases, it can be shewn that if the gas composing the stream expands adiabatically and all the work done is expended

in increasing the kinetic energy of the stream, then the velocity of the latter will increase as expansion proceeds and attain a maximum value given by : —

$$V = \sqrt{2g\,J\,K_p\,(T_1 - T_2)} = 109{\cdot}5\sqrt{(T_1 - T_2)} \text{——}(1)$$

where :—

V = maximum velocity of stream in ft. /sec.
g = 32·2 ft. /sec.² (acceleration due to gravity).
J = Joules' equivalent (778 ft. lb. /B.T.U.).
K_p = Specific heat at constant pressure (for air K_p = ·238 B.T.U. /lb. deg. F.
T_2 = Temperature attained by the stream after adiabatic expansion from P_1 to P_2, and is given (for air) by :—

$$T_2 = T_1\left(\frac{P_2}{P_1}\right)^{0.285} \text{——}(2)$$

In equation (1) the velocity in the chamber is supposed to be negligibly small compared with V. For values of P_1 up to

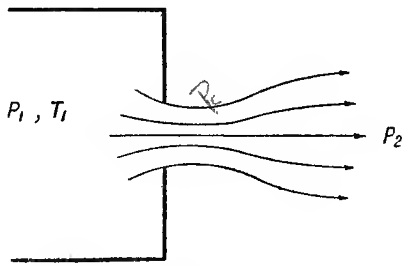


Fig. 12.

70 × 144 lb. per sq. ft., P_2 being atmospheric, equation (1) has been found to agree with experiment within two or three per cent, which is ample accuracy for our purpose.

It may, however, be mentioned in passing that experiments at high pressures, and particularly with steam, indicate that the elementary theory on which equation (1) is based stands in need of correction.

It is to be observed that equation (1) is equally valid for any stage of the expansion of the stream, so that if any pressure value P_A be selected lying between P_1 and P_2, then the velocity at this stage will be given by equation (1) with P_A substituted for P_2. If values of the velocity V_A be calculated for various values of P_A and the specific volumes v_A corre-

sponding to these values, then the ratios $\frac{v_A}{V_A}$ will be proportional to the areas of the stream at the several stages of expansion. On making such a calculation it will be found that the area of the stream at first contracts and afterwards converges to a final value corresponding to the maximum velocity. In other words, the stream has a neck or throat, as shown in Fig. 12. The pressure P_C at the throat is known as the critical pressure, and for air is equal to $0{\cdot}53\ P_1$.

Furthermore, it is evident that if P_2 is equal to P_C (a contingency which was ruled out at the beginning of the discussion), then the stream will converge to its throat area and remain parallel instead of diverging.

Supposing again that $P_2 < P_C$, it is evident that for a given size of orifice the discharge will be a maximum if the throat of the stream occurs at the orifice, and in this case the discharge in lb. per sec. will be given by :—

$$Q = A\ V_C\ W_C \text{——}(3)$$

where Q = discharge in lb. per sec.
A = area of orifice—ft.^{2}
V_C = throat velocity—ft./sec.
W_C = weight in lb. per cub. ft. of the air at the conditions of temperature and pressure obtaining at the throat.

So that the discharge is independent of the back pressure P_2 so long as $P_2 < P_C$.

On the other hand, there is no guarantee that the throat of the stream will coincide with the orifice in every case, so that in general the discharge given by equation (3) has to be multiplied by a discharge coefficient less than unity, in order to give the discharge observed by experiment.

For air and exhaust gases the following figures may be used :—

Discharge coefficient for sharp-edged orifices or ports	0·65
,, ,, ,, mushroom valves . . .	0·70

The velocity calculated from equation (1), multiplied by the appropriate coefficient, may conveniently be called the apparent velocity referred to the actual area of the orifice.

The Suction Stroke.—By way of application of the preceding formulæ, we may consider the suction stroke of a Diesel Engine. The retreating piston creates a partial vacuum and air passes

in from the atmosphere via the inlet valve in consequence. Supposing it is desired to limit the pressure difference to 1 lb. per sq. in., then—

$$P_1 = 14{\cdot}7 \times 144 \quad P_2 = 13{\cdot}7 \times 144 \quad T_1 = \text{say } 520^\circ \text{ F. abs.}$$

$$\text{Then } T_2 = 520\left(\frac{13{\cdot}7}{14{\cdot}7}\right)^{0{\cdot}285} = 509{\cdot}7^\circ \text{ F. abs.}$$

$$\text{And } V = 109{\cdot}5\sqrt{520 - 509{\cdot}7} = 351 \text{ ft. /sec.}$$

Using a coefficient of 0·7 for the value, the apparent velocity referred to the valve area is

$$351 \times 0{\cdot}7 = 246 \text{ ft. /sec.}$$

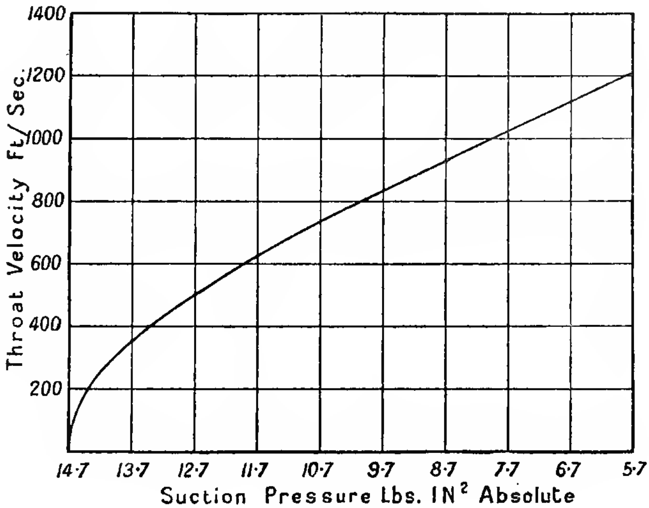


Fig. 13.

If the pressure difference is to be constant at 1 lb. /in.² then the valve and its operating cam must be so designed that

$$\frac{\text{Instantaneous piston speed (ft. /sec.)}}{246} = \frac{\text{Piston area}}{\text{Instantaneous valve area.}}$$

As indicated in Chapter I, it is better to allow the inlet valve to open before the inner dead centre, and in order to obtain a maximum charge of air the valve should not seat until 20° or 30° after the outer dead centre has been passed.

Fig. 13, which is a curve connecting calculated velocity and suction pressure, is the result of repeating the above for different values of P_2 on the assumption that

$$P_1 = 14{\cdot}7 \times 144 \text{ and } T_1 = 520^\circ \text{ F. abs.}$$

Scavenging of Two Stroke Cylinders.—Before the scavenge air supply is put into communication with the cylinder, whether by valves or ports, the exhaust slots should have been sufficiently uncovered for the cylinder pressure to have fallen to practically atmospheric pressure. As will be shown later, this takes place with great rapidity owing to the high temperature of the gases. On this account the introduction of scavenge air may be assumed to take place against a pressure differing but very slightly from atmospheric. This, however, does not apply to the later stage of the process in those valve scavenged or controlled port scavenge engines in which scavenge air is

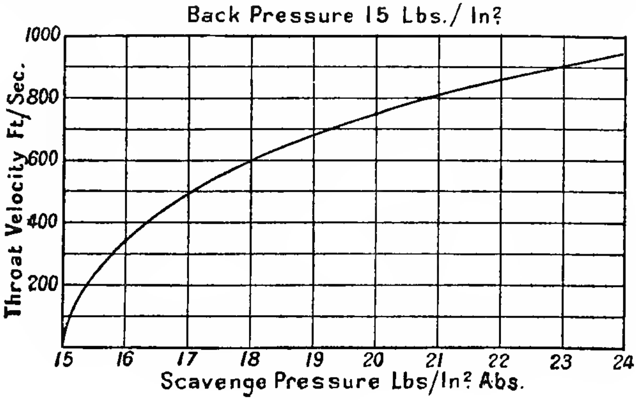


FIG. 14.

forced into the cylinder after the exhaust ports have been covered. During this latter stage the cylinder pressure continually increases until either the scavenge valves (or ports) close and compression begins, or until the cylinder contents and the scavenge air supply are in equilibrium.

The first stage of the process may be dealt with in a similar manner to that indicated in the preceding article, and Fig. 14 shows the calculated throat velocity of the scavenge air plotted against the absolute scavenge pressure, on the assumption that the back pressure is 15 lb. per sq. in. abs. and the temperature of the scavenge air supply is 130° F. (591° F. abs.), this being a good average figure found in practice.

Before applying the above to a concrete example some observations will be made on port and valve areas.

If a fluid flows for a time " t " through an orifice of constant

area "A," with a velocity "V," then the volume "v" discharged is evidently given by :—

$$v = V\,(A \times t)$$

If on the other hand A is variable, then

$$v = V\int A \cdot dt$$

The quantity $\int A \cdot dt$ known as the time integral of the area is readily found by plotting A as ordinate against "t" as abscissæ, as in Fig. 15, which exhibits the opening area of a valve from the time it lifts ($t = o$) to the time it seats.

The time integral of the valve area for the whole interval is the area under the curve. For any other interval from, say, $t = t_1$ to $t = t_2$ the time integral of the area is the area bounded by curve, the axis of t and the two ordinates which define

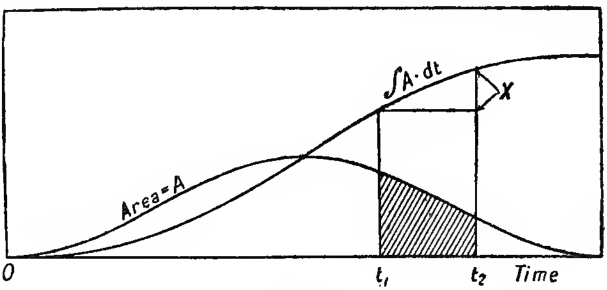


FIG. 15.

t_1 and t_2. This area is shown shaded on the diagram for particular values of t_1 and t_2.

It is often convenient to take as the unit for "t" $\frac{1}{100}$th sec. or $\frac{1}{1000}$th sec., or even 1 degree of revolution of the crank-shaft.

Also it is very convenient to plot on a "t" base a curve whose ordinates represent $\int A \cdot dt$ from $t = o$.

This has been done on Fig. 15. It will be noticed that between the instants $t = t_1$ and $t = t_2$ the value of $\int A \cdot dt$ has increased by an amount X. The volume discharged between these instants would therefore be $(X \cdot V)$.

Example of Scavenge Calculation.—Data for two stroke engine :—

Bore of cylinder	10″
Stroke	15″
Revolution per minute . . .	300

Two scavenge valves $3\frac{1}{2}''$ diam., maximum lift $1''$, opening 25° before bottom dead centre and closing 60° after. Lift curve harmonic, i.e. the lift plotted on a time base is a sine curve. Exhaust ports occupy 60% of the circumference of the cylinder bore and become uncovered by the piston 15% before the end of the stroke. The connecting rod is 5 cranks long. Fig. 16 has been drawn to show the relation between percentage of

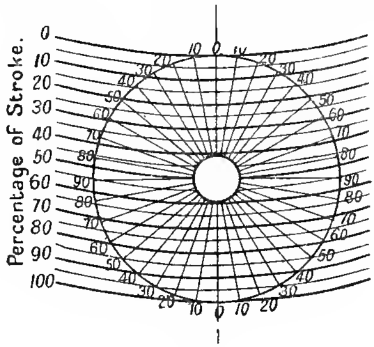


FIG. 16.

stroke and the number of degrees between the crank position and the bottom dead centre. From this diagram it will be seen that the ports are uncovered 50° before B.D.C. and covered again 50° after B.D.C. Compression space 7% of the stroke volume. Free air capacity of scavenger 50% in excess of stroke volume of impulse cylinders.

$$\text{Stroke volume} = \frac{\cdot 785 \times 10^2 \times 15}{1728} = 0\cdot 68 \text{ ft.}^3$$

$$\text{Compression space} = 0\cdot 07 \times 0\cdot 68 = 0\cdot 05 \text{ ,,}$$

$$\text{Stroke volume} + \text{compression space} = 0\cdot 73 \text{ ,,}$$

Volume of scavenge air (at atmospheric pressure and temperature) delivered per cylinder per revolution $= 0\cdot 68 \times 1\cdot 5 = 1\cdot 02$,,

Maximum exhaust port area

$$= \frac{0\cdot 6 \times \pi \times 10 \times 0\cdot 15 \times 15}{144} = 0\cdot 295 \text{ ,,}$$

Maximum scavenge valve area

$$= \frac{2 \times \pi \times 3\cdot 5 \times 1}{144} = 0\cdot 153 \text{ ft.}^2$$

Intermediate values of the exhaust port and scavenge valve areas have been plotted on a crank angle base on Fig. 17.

Number of seconds corresponding to 1 degree of revolution of the crank-shaft

$$=\frac{60}{300}\times\frac{1}{360}=0{\cdot}000556 \text{ secs.}$$

In Fig. 17 values of $\int A{\cdot}dt$ for both exhaust and scavenge areas have been plotted in accordance with the preceding article. In particular, since the scavenge valve lift is harmonic,

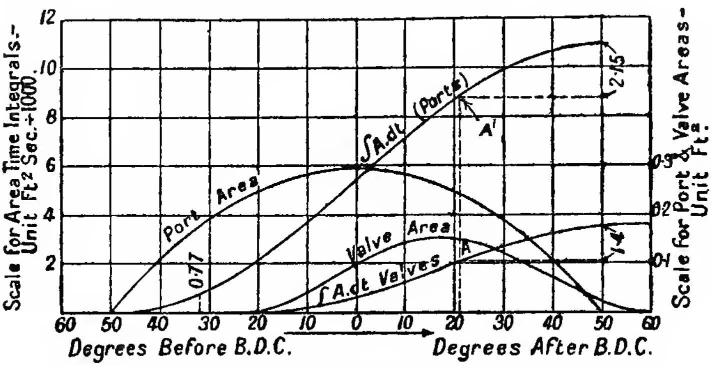


Fig. 17.

the mean area is one-half the maximum, and the final value of $\int A{\cdot}dt$ is given by :—

$$\frac{0{\cdot}153}{2}\times 0{\cdot}000556\times 85^{\circ}=10^{-3}\times 3{\cdot}61 \text{ ft.}^{2} \text{ secs.}$$

And since the port area curve is nearly parabolic the maximum value of $\int A{\cdot}dt$ for the ports is given by :—

$$0{\cdot}295\times\tfrac{2}{3}\times 0{\cdot}000556\times 100^{\circ}=10^{-3}\times 10{\cdot}94 \text{ ft.}^{3} \text{ secs.}$$

The intermediate values are readily found by planimetering the areas under the port and valve area curves.

Scavenge Air Pressure.—A first approximation to the value of the pressure in the scavenger air pipe required by the conditions of the problem is readily obtained, on the assumption that the scavenge air is delivered against a constant pressure of 15 lb. per sq. in. abs.

Volume discharged . . . $=1{\cdot}02$ ft.^{3}

$\int A{\cdot}dt$ . . . . . . $=3{\cdot}61\times 10^{-3}$

Apparent velocity . . . $=\frac{1{\cdot}02\times10^{-3}}{3{\cdot}61}=283$ ft. /sec.

Dividing by a discharge coefficient of 0·7 the "calculated" velocity is $283\div0{\cdot}7=405$ ft. /sec.

and the corresponding scavenge air pressure from Fig. 14 is :—

$$=16{\cdot}35 \text{ lb. per in.}^2 \text{ abs.}$$

This figure cannot, however, be accepted as final, since the effect of discharging the excess of air through the exhaust ports has been ignored.

Assume for trial a scavenge pressure of 17 lb. per in.^{2} abs. The calculated velocity from Fig. 14 is 498 ft. /sec. and the apparent velocity

$$=498\times0{\cdot}7=349 \text{ ft. /sec.}$$

The value of $\int A{\cdot}dt$ which must be attained to fill the stroke volume and clearance space (0·73 ft.^{3}) is therefore :—

$$\frac{0{\cdot}73}{349}=2{\cdot}09\times10^{-3}$$

This corresponds to point A on Fig. 17, and shows that with the assumed scavenge pressure the cylinder would be filled with scavenge air at about 21° after the bottom dead centre. From this point until the point where the exhaust ports are covered three processes are occurring simultaneously, viz. :—

(1) Scavenge air is escaping out of the exhaust ports.
(2) Scavenge air is entering the cylinder and raising the pressure of its contents.
(3) The piston is rising and reducing the volume of the cylinder contents, at the same time tending to raise the pressure.

The pressure which exists in the cylinder at the instant at which the piston covers the ports may conveniently be termed the "initial charge pressure," and will be denoted by P_i. Now if the value of P_i were equal to the scavenge pressure, which is its upper limit, the amount of air introduced into the cylinder in the interval under consideration would be about equal to :—

$$\tfrac{2}{3}\int A{\cdot}dt \text{ (from 21° to 50° after B.D.C.)}\times349$$
(for valves)

From Fig. 17 $\int A \cdot dt$ for this interval is $1{\cdot}4 \times 10^{-3}$, so the volume introduced is

$$\tfrac{2}{3} \times 1{\cdot}4 \times 10^{-3} \times 349 = 0{\cdot}326 \text{ ft.}^3$$

Adding to this the volume already introduced, viz. 0·73 ft.^{3}, we obtain 1·056 ft.^{3}, which is a trifle in excess of the required quantity.

The value assumed for the scavenge pressure, viz. 17 lb. /in.^{2} abs., is, therefore, probably not far out.

The precise value of P_1 is a matter of considerable importance, as on it depends the value of the charge weight, and, consequently, the power capacity of the cylinder. Its predetermination, however, appears to be a difficult matter, and in practice it is usually adjusted experimentally by advancing or retarding the scavenge cam, according as the value of P_i found by light spring indicator diagram is too low or too high.

In the former case an adjustment is easily effected by putting a baffling diaphragm in the exhaust pipe.

The above calculations, however, are a sufficient check on the design to ensure the provision of suitable valve or port areas.

The volume (@ 15 lb. /in.^{2} abs.) of scavenge air lost through the exhaust ports is roughly equal to

$$\tfrac{1}{3} \int A \cdot dt \text{ (from } 21^\circ \text{ to } 50^\circ \text{ after B.D.C.)} \times 498 \times 0{\cdot}65$$
$$\text{(for ports)}$$

From Fig. 17 $\int A \cdot dt$ for this interval is $2{\cdot}15 \times 10^{-3}$ ft.^{2} sec., so the volume required is

$$\tfrac{1}{3} \times 2{\cdot}15 \times 10^{-3} \times 498 \times 0{\cdot}65 = 0{\cdot}23 \text{ ft.}^3$$

So that the air charge is equivalent to $(1{\cdot}02 - 0{\cdot}23) = 0{\cdot}79$ ft.^{3} @ 15 lb. per sq. in. abs.

Its actual volume at the point when the piston covers the ports is $0{\cdot}05 + (0{\cdot}85 \times 0{\cdot}68) = 0{\cdot}63$ ft.^{3}

and its pressure is therefore about

$$\frac{15 \times 0{\cdot}79}{0{\cdot}63} = 18{\cdot}8 \text{ lb. /in.}^2$$

This figure being 1·8 lb. /in.^{2} above the assumed scavenge pressure of 17 lb. /in.^{2} indicates that the latter figure is insufficient for the supply of the requisite quantity of air under the conditions specified. The true value probably lies somewhere between the two.

Exhaust of Two Stroke Engines.—As already mentioned,

it is essential that the scavenge receiver should not be put into communication with the cylinder until the contents of the latter have fallen to a pressure almost equal to that of the scavenge air, by the release of the products of combustion, through the exhaust slots. This process usually takes a period of time equivalent to 20 to 30 degrees of revolution of the crank-shaft. The calculation of this period is much facilitated by the fact that in the interval considered the port area is increasing very approximately in proportion to the time (see Fig. 17). It can easily be shown that if during an interval from $t=o$ to $t=t_1$ an orifice area increases uniformly with respect to time from O to A_1 and the velocity of efflux also varies uniformly with respect to time from V_0 to V_1, then the discharge is given by :—

$$Q=\frac{A_1T_1}{3}(V_1+\tfrac{1}{2}V_0) \text{——(1)}$$

If A represents ft.^{2}, t secs., and V ft./secs. of an incompressible fluid, then Q represents cub. ft.

If, as in the case we are about to consider, V represents lb. per sec. per unit of area, then Q represents lb. We proceed to apply equation (1) to the exhaust period of the two stroke engine specified in the previous article.

Exhaust Calculation.—*Data :* Cylinder pressure at the point at which the ports become uncovered, 60 lb./in.^{2} abs. Temperature at the same point is equal to

$$\frac{\text{initial charge temperature}\times 60}{\text{,, \quad ,, \quad pressure.}}=\text{say}\ \frac{670\times 60}{18}$$
$$=2240^\circ \text{ F. abs.}$$

Pressure in exhaust pipe, 15 lb./in.^{2} abs.

Problem : To find the position of the crank when the cylinder pressure has fallen to 18 lb./in.^{2} abs.

Assume that the charge has the thermal properties of pure air and that the expansion is adiabatic.

At the instant when the ports first become open the volume of the charge is :—

$$0{\cdot}05+0{\cdot}85\times 0{\cdot}68=0{\cdot}63 \text{ ft.}^3$$

and the weight of the charge is :—

$$(60\times 144\times 0{\cdot}63)\div(53{\cdot}2\times 2240)=0{\cdot}0456 \text{ lb.}$$

When the cylinder pressure has fallen to 18 lb./in.^{2} abs. the

volume is not certainly known, but will not differ much from $0{\cdot}05+0{\cdot}92\times0{\cdot}68=0{\cdot}675$ ft.^{3}, and its temperature will be :—

$$2240\left(\frac{18{\cdot}0}{60{\cdot}0}\right)^{\cdot285}=1590^\circ \text{ F. abs.}$$

and the charge weight is reduced to

$$(18\times144\times0{\cdot}675)\div(53{\cdot}2\times1590)=0{\cdot}0207 \text{ lb.}$$

The weight discharged in the interval is therefore :—

$$0{\cdot}0456-0{\cdot}0207=0{\cdot}0249 \text{ lb.}$$

The next step is to calculate the rate of discharge per unit of port area.

At the higher pressure of 60 lb./in.^{2} the throat pressure is $0{\cdot}53\times60=31{\cdot}8$ lb./in.^{2} abs., and the throat temperature is :—

$$2240\left(\frac{31{\cdot}8}{60}\right)^{\cdot285}=1870^\circ \text{ F. abs.}$$

The calculated throat velocity is therefore :—

$$109{\cdot}5\sqrt{2240-1870}=2110 \text{ ft./sec.}$$

and the apparent velocity $=2100\times0{\cdot}65=1365$ ft./sec.

Now the specific volume at the throat is :—

$$\frac{53{\cdot}2\times1870}{31{\cdot}8\times144}=21{\cdot}8 \text{ ft.}^3 \text{ per lb.}$$

The rate of discharge is therefore :—

$$\frac{1365}{21{\cdot}8}=62{\cdot}7 \text{ lb. per sec. per ft.}^2 \text{ of port area.}$$

At the lower pressure of 18 lb./in.^{2} the throat pressure is 15 lb./in.^{2} abs., and the throat temperature is :—

$$2240\left(\frac{15}{60}\right)^{\cdot285}=1510^\circ \text{ F. abs.}$$

The calculated throat velocity is therefore :—

$$109{\cdot}5\sqrt{1590-1510}=978 \text{ ft./sec.}$$

and the apparent velocity $=978\times0{\cdot}65=635$ ft./sec.

Now the specific volume at the throat is :—

$$\frac{53{\cdot}2\times1510}{15\times144}=37{\cdot}2 \text{ ft.}^3 \text{ per lb.}$$

and the rate of discharge is therefore :—

$$\frac{635}{37{\cdot}2}=17{\cdot}1 \text{ lb./sec. per ft.}^2 \text{ of port area.}$$

We now make the assumption that the rate of discharge per unit area changes its value from the higher to the lower value uniformly with respect to time, so that equation (1) can be applied. We then have :—

$$0{\cdot}0249=\frac{A_1\,t_1}{3}\left(17{\cdot}1+\frac{62{\cdot}7}{2}\right)$$

from which $A_1\,t_1=10^{-3}\times 1{\cdot}54$

and $\int A{\cdot}dt=\frac{1}{2}\,A_1\,t_1$ (approx.) $=10^{-3}\times 0{\cdot}77$ ft.^{2} sec.

Reference to Fig. 17 shows that this value occurs about 18 degrees after the ports begin to open.

Alternative Method of Calculation.—

Let :—

t = time in seconds, counting from the instant when the ports begin to be uncovered by the piston.

P = pressure of cylinder contents in lb./ft.^{2} during the period considered, so that P varies from 60×144 to 18×144.

T = temperature of cylinder contents deg. F. abs. (variable).

w = weight in lb. of cylinder contents (variable).

A = port area in ft.^{2} (variable).

v = volume in ft.^{3} of cylinder contents (actually varies during the period considered, but treated as constant) $=0{\cdot}65$ ft.^{3}

v_s = specific volume in ft.^{3}/lb. at the throat of the stream issuing from the ports.

V = throat velocity in ft./sec.

$f(P)=\frac{dP}{dt}\div A=$ the rate of pressure drop per unit of port area,

then $\frac{dP}{dt}=f(P)\times A$

and $\int A{\cdot}dt=\int\frac{dP}{f(P)}$

f(P) has now to be calculated.

$$P.v=53{\cdot}2\;wT \qquad \therefore\quad P=\frac{53{\cdot}2}{v}w.T.$$

$$\text{and}\quad \frac{dP}{dt}=\frac{53{\cdot}2}{v}\left(T.\frac{dw}{dt}+w.\frac{dT}{dt}\right)$$

$$=\frac{53{\cdot}2}{v}\left(T.\frac{dw}{dt}+w.\frac{dT}{dP}.\frac{dP}{dt}\right)$$

so that

$$\frac{dP}{dt}=\frac{\frac{53\cdot 2}{v}\left(T.\frac{dw}{dt}\right)}{\left(1-\frac{53\cdot 2}{v}.w.\frac{dT}{dP}\right)}=\frac{\frac{53\cdot 2}{v}.T.\frac{dw}{dt}}{\left(1-\frac{p}{T}.\frac{dT}{dP}\right)}$$

Now $\frac{dw}{dt}=A.\frac{V}{v_s}$

and since $T=T_o\left(\frac{P}{P_o}\right)^{0\cdot 285}$, where T_o and P_o are the initial values of T and P,

$$\frac{dT}{dP}=\frac{0\cdot 285\ T_o P^{(0\cdot 285-1)}}{P_o^{0\cdot 285}}$$

and $\frac{P}{T}\cdot\frac{dT}{dP}=0\cdot 285$ (i.e. constant),

so that $f(P)=\frac{dP}{dt}\div A=\frac{\frac{53\cdot 2}{v}.T.\frac{V}{v_s}}{0\cdot 715}$

Values of f(P) are easily calculated for evenly spaced values of P and the integration of $\int\frac{dP}{f(P)}$ can be effected by Simpson's rule.

The table opposite shows the application of this method to the example of the preceding article.

$P_o=60$ lb./in.^{2} abs. $T_o=2240°$ F. abs. $V=0\cdot 65$ ft.^{3}

For convenience P has been expressed in lb./in.^{2}, instead of lb./ft.^{2}, so that a factor of 144 is required. Since the interval between successive values of P is 10·5 lb./in.^{2}, we have :—

$$\int A\cdot dt=\frac{dP}{f(P)}=\frac{10\cdot 5\times 144\times 96\cdot 18\times 10^{-8}}{3}$$

$$=0\cdot 485\times 10^{-3}$$

Dividing by the discharge coefficient 0·65 the required value of $\int A\cdot dt$ is :—

$$\frac{0\cdot 485\times 10^{-3}}{0\cdot 65}=0\cdot 74\ \text{ft.}^2\ \text{sec.}\div 1000$$

which agrees with the value obtained previously within about 4%.

The value of the discharge coefficient (0·65) has been found by comparing the results of calculations similar to the above, with the information afforded by light spring indicator cards.

1	2	3	4	5	6	7	8	9	10	11	12
P lb./in.² abs.	T deg. F. abs.	Throat Press. lb./in.² abs.	Throat Temp. deg. F. abs.	Temp. drop, deg. F.	Throat velocity, V ft./sec.	v_s	$\frac{V}{v_s}$	$\frac{53 \cdot 2\, T \cdot V}{v \cdot v_s}$	$\frac{1}{f(P)} = \frac{0 \cdot 715}{[9]}$	Simpson's figure.	(10)×(11)
60·0	2240	31·7	1870	370	2110	21·8	96·8	$1 \cdot 78 \times 10^7$	$4 \cdot 02 \times 10^{-8}$	1	$4 \cdot 02 \times 10^{-8}$
49·5	2120	26·2	1770	350	2050	25·1	81·7	$1 \cdot 42 \times 10^7$	$5 \cdot 04 \times 10^{-8}$	4	$20 \cdot 16 \times 10^{-8}$
39·0	1980	20·6	1655	325	1975	29·8	66·4	$1 \cdot 08 \times 10^7$	$6 \cdot 62 \times 10^{-8}$	2	$13 \cdot 24 \times 10^{-8}$
28·5	1810	15·0	1510	300	1900	37·2	51·1	$0 \cdot 76 \times 10^7$	$9 \cdot 44 \times 10^{-8}$	4	$37 \cdot 76 \times 10^{-8}$
18	1590	15·0	1510	80	980	37·2	26·3	$0 \cdot 34 \times 10^7$	$21 \cdot 00 \times 10^{-8}$	1	$21 \cdot 00 \times 10^{-8}$
											$96 \cdot 18 \times 10^{-8}$

It is also in good agreement with experiments on the flow of gases through sharp-edged orifices. It is by no means easy, however, to obtain light spring indicator diagrams which are at all reliable with an ordinary indicator. The fall of pressure is so rapid that the shape of the card is distorted by very little indicator stickiness, and oscillations are almost inevitable.

With care the latter may be approximately allowed for, but cards showing appreciable indicator stickiness must be rejected. These troubles may be largely eliminated by using an optical indicator, which is perhaps the only instrument well adapted to this class of investigation.

Fig. 18 shows a light spring diagram taken from a two stroke Diesel Engine with an ordinary indicator in good order.

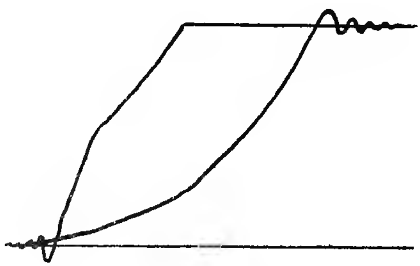


FIG. 18.

Literature.—Funck, G., "Two Stroke Engines with special reference to the Design and Calculation of Ports."—*Automobile Engineering*, May, 1918, *et seq.*

Petter, H., "The Escape of Exhaust Gas in Two Stroke Engines."—*Engineering*, January 4th, 1918.

Thompson Clarke Research, "Air Flow through Poppet Valves."—Report to U.S. Nat. Advisory Committee for Aeronautics. See *Engineering*, January 10th, 1919.

CHAPTER IV

THE PRINCIPLE OF SIMILITUDE

THE properties of similar structures, under equivalent conditions, have been known for a long time, and are implied in most of the empirical formulæ used in machine design. At the same time, there is a great deal of misconception about the matter in the minds of many to whom a correct appreciation of the principles of similitude would be of considerable value.

Definition of Similar Engines.—For the purposes of this discussion, two or more engines are said to be similar when the linear dimensions of every part of one engine bear a constant ratio to the linear dimensions of the corresponding parts of the other engine or engines. Stating the same condition in another way, any dimension of any part (the diameter of the gudgeon pin, for example) can be expressed as a fraction or proportion of some other dimension (the cylinder bore, for example), this fraction or proportion being constant for all similar engines.

It follows from the above that two engines to be similar must have the same bore to stroke ratio.

In practice there are a considerable number of deviations from strict similarity between different sized engines of the same type, built by the same maker, a few being noted below :—

1. The bore or stroke ratio is subject to variation.
2. Thicknesses of metal generally bear a larger fractional ratio to the cylinder bore in small engines than in larger engines.
3. Studs, bolts and other small gear are made relatively heavier in the smaller sizes, to avoid damage by careless handling, etc.

Equivalent Conditions.—Similar engines may be said to be under "equivalent conditions" when the piston speed is the same and the indicator card identical for both engines.

In the past it has been customary to reserve the higher piston speeds for the larger sizes of engines; the modern tendency, on the other hand, is to use the same piston speed for all sizes. The former practice appears to have secured uniform durability as measured by the useful life of the engine for all sizes, whereas the latter undoubtedly results in the smaller engines being subject to more rapid depreciation than the larger. Without appreciable error the indicator cards, so far as they influence the stresses in the various component parts of the engine, may be taken as identical for all sizes, though it is worth mentioning here that it is found advisable in practice to work with a smaller M.I.P. in the case of large cylinders.

Properties of Similar Engines.—To avoid repetition equivalent conditions will be implied when use is made of the term "similar engines," unless the contrary is specified. Since all parts of similar engines are in proportion their relative size may be expressed by any linear dimension of any part, and for this purpose it is very convenient to select the cylinder bore, as this is the dimension which, assuming constant piston speed, determines the power of the cylinder. Consider a whole series of similar engines having cylinders of different bores. Now the piston load at any point of the stroke will be proportional to the bore^2, since the indicator card is the same for all. Now the area of the main bearings will also be proportional to the bore^2 since both linear dimensions contributing to the bearing area are proportional to the bore. From this it follows that the bearing pressures are the same for all the engines. Without detailing the matter further, it suffices to state that with similar engines all the bearing pressures and stresses (including inertia stresses) of corresponding parts are identical. Dealing in the same way with rubbing speeds of bearings, velocities of gases, etc., it is found that these also are identical. From these facts it follows that given a satisfactory engine other engines made from the same designs but to a different scale (within rational limits) will also be workable machines, though for practical reasons such a procedure carried out in minutiæ would not be desirable. Actually, the foremost makers of Diesel Engines have carried out the principle of similarity remarkably closely.

Relative Weight of similar Engines.—Weight being proportional to volume (assuming the same materials are always used for corresponding parts), the weights of similar engines are

proportional to the bore^3. Hence in comparing two different constructions of engine the weight per cubic inch of bore^3 is a convenient criterion of the heaviness of construction. Figures for actual engines will be given later. Again, since the power varies as the bore^2 the weight per horse-power varies as the bore with similar engines. Owing to departures from strict similarity, some of which have been mentioned above, the weight per horse-power does not in practice increase proportionally as the bore is increased. The practice of adopting higher piston speeds for the larger size of engine tends in the same direction, with the result that the weight per B.H.P. shews comparatively small variation over a large range of powers.

Circumstances which act in the reverse direction are the necessity for crossheads and guides and slight reduction of M.I.P. in the larger sizes.

In comparing the weight per B.H.P. of Land Diesel Engines the number of cylinders must be taken into consideration for two reasons :—

1. The cam-shaft driving gear is usually the same for a six or four cylinder engine as for a single cylinder engine of the same bore and type, and consequently bears a larger proportion to the total weight in the case of the engine with the smaller number of cylinders. The same applies to most of the accessories, such as starting bottles, fuel filters, etc., if these are included in the weight.
2. The weight of the fly-wheel is generally less for a three cylinder engine than for a single cylinder, the bore being the same, though it will be shewn later that further reduction of fly-wheel weight is not advisable for four and six cylinders.

A summary of the properties of theoretically similar engines is given below.

Piston speed constant. M.I.P. constant.

1. Linear dimensions proportional to the bore.
2. Revolutions per minute inversely as the bore.
3. Rubbing velocities and gas velocities are the same.
4. Bearing pressures are the same.
5. Stresses are the same (including the inertia stresses).

6. Elastic deflections proportional to the bore.
7. Natural frequencies of vibration proportional to the R.P.M.
8. Loads (pressure and inertia) proportional to the bore^2.
9. Horse-power proportional to the bore^2.
10. Weight proportional to the bore^3.
11. Weight per B.H.P. proportional to the bore.
12. Weight per unit of bore^3 the same.

The above considerations justify to a great extent the common practice of comparing different designs by expressing

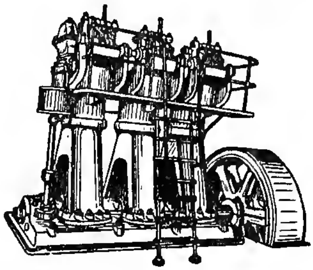


FIG. 19. FOUR STROKE LAND ENGINE.

the dimensions of corresponding parts as fractions of the bore regardless of whether machines compared are of the same approximate size or not. The remainder of this chapter will be devoted to applications.

Weight of Diesel Engines.—From the purchaser's point of view the weight of a power installation per B.H.P. is a matter of some importance, and Fig. 20 shews how this quantity varies with different sizes of four stroke land engines of the well-known "A" frame type. It will be observed that the weight per B.H.P. increases with the size of the cylinder, but not as rapidly as the principle of similitude would lead one to expect. This is due in part to the fact that the weights include a quantity of auxiliary gear, such as air-bottles, etc., which form a larger percentage of the whole weight in the case of smaller engines. The fact of the weight per B.H.P. varying considerably in different sizes renders this figure very unsuitable for estimating purposes. The table below shews the weight per cubic inch of bore^3 for the same range of engines, and although here also the influence of the weight of the auxiliary gear may

be noticed there is a much better approximation to constancy in the figures.

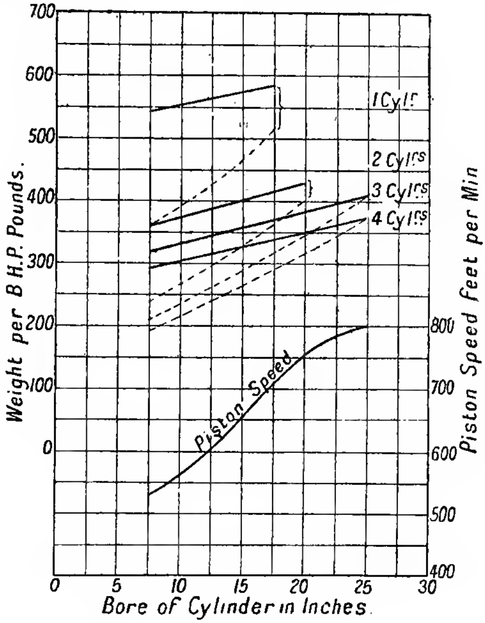


FIG. 20.

TABLE I

Weight in lb. ÷ [Bore³(inches) × No. of Cylinders.]

1 Cylinder.		2 Cylinders.		3 Cylinders.		4 Cylinders.	
Bore.	lb.	Bore.	lb.	Bore.	lb.	Bore.	lb.
9·0	15·0	9·0	12·0	12·6	7·1	16·5	6·2
10·6	13·2	10·6	8·5	13·8	7·1	18·1	6·1
11·4	12·0	11·4	8·3	15·0	7·1	22·4	6·1
12·6	11·2	12·6	7·8	18·1	6·7	—	—
13·8	11·2	13·8	7·8	20·5	6·5	—	—
15·0	11·2	15·0	7·8	—	—	—	—

Table II below gives similar figures deduced from the published weights of a line of four stroke land engines of the

crank-case type with trunk pistons, and all having three cylinders. In this case the fly-wheel and auxiliary gear are not included.

TABLE II

Bore in in.	10·25	12	14	16
Stroke in in.	15	18	21	24
Weight in lb. (bore$^3 \times 3$)	5·7	5·2	6·6	5·2

From these and other figures it appears that the crank-case construction is not intrinsically much lighter than the "A" frame type of construction, and that with the former any economic advantages are traceable to the higher piston speeds which become practicable when the crank-case is enclosed.

Similar figures are given in Table III for various types of Marine Diesel Engines. The figures include the fly-wheel and piping attached to the engine, but no auxiliary gear.

TABLE III

Description.	Weight in lb. per in.^{3} of bore^3 × No. of cylind's.
Large two stroke. Cast iron columns similar to steam engine practice; water pumps included	9·5 to 14
Large two stroke. Structure of trestle type with through bolts; water pumps included	5·2
Large four stroke structure of trestle or crank-case type with through bolts only, the H.P. stage of compressor included	4·5
Large four stroke. Structure of stay-bolt type	4·0
Trunk engines of crank-case type	4·8
Trunk engines of stay-bolt type	3·5

For preliminary estimating it is convenient to have an approximate idea of the weights of the piston, connecting rod, etc., of a proposed engine, and the following figures are a rough guide to average practice. A designer will find it convenient to make notes of similar figures for actual engines with which he has had experience.

TABLE IV

Name of Part.	Weight per in.^{3} of bore^3 lb.
Trunk piston (land engines)	0·18
Connecting rod ,, ,,	0·17
Crank-pin ,, ,,	0·036
One web ,, ,,	0·036
Reciprocating weight (land engines)	0·24
Revolving weight ,,	0·18
Piston and piston rod (marine engine)	0·13
Crosshead ,, ,,	0·13
Connecting rod ,, ,,	0·24
Crank-pin ,, ,,	0·046
One web ,, ,,	0·046
Reciprocating weight ,, ,,	0·355
Revolving weight ,, ,,	0·236

The above figures are necessarily approximate only, as they depend not only on the materials used and the views as to design stresses, etc., held by individual designers, but also on the bore to stroke ratio and the rules of insurance societies.

Determination of Bore and Stroke.—Nowadays Diesel Engines are generally built up of standardised units in combinations of 2, 3, 4, 6 and 8 cylinders, and by this means a large range of sizes may be covered with a minimum of stock patterns, jigs, etc. The choice of sizes of cylinders and the numbers of different sizes stocked will of course depend on the capacity of the factory and the estimated demand. Marine engines are usually provided with 6 or 8 cylinders in the case of four stroke engines and 4 or 6 cylinders in the case of two stroke engines, to facilitate starting and manœuvring and also to keep the size of cylinder and the aggregate weight within reasonable limits. Having decided on the brake-power to be developed by a certain standard cylinder the indicated power must be inferred, either from previous experience of engines similar to the one proposed, by direct estimate of the various losses, or with reference to published data of comparable engines. Typical figures for mechanical efficiency have been given on page 27.

A very close approximation of the mechanical losses is obtainable as follows :—

1. Air compressor and scavenger losses may be reckoned to be equal to the indicated power of these accessories divided by a mechanical efficiency of 0·8.
2. Losses due to friction of piston rings about 5% of the indicated power of the engine.
3. Lubricated friction loss is given very approximately by the formula :

Horse-power lost in lubricated friction

$$=\frac{0{\cdot}3\,[D.L_1V_1{}^{1\cdot5}+n(A+BL_2)V_2{}^{1\cdot5}]}{550}$$

Where

D = Diameter of crank-shaft in in.
L_1 = Aggregate length of journals and crank-pins in in.
V_1 = Peripheral speed of journals in ft. per sec.
n = Number of cylinders.
A = Area of slipper guide in sq. in.
B = Bore of cylinder in in.
L_2 = Length of piston in in.
V_2 = Piston speed in ft. per sec.

Assuming that the desired indicated power per cylinder is now known, the former is related to the cylinder bore by the formula :—

$$\text{I.H.P. per cylinder}=\frac{0{\cdot}785.B^2P.V}{2\times 33{,}000}=\frac{B^2.P.V}{84{,}000} \qquad (1)$$

for two cycle engines.

And half this amount for four cycle engines, both assumed single acting.

Where

P = Mean indicated pressure in lb. per sq. in.
V = Piston speed in ft. per minute.

The piston speed and mean indicated pressure both vary considerably in different cases, and existing practice in this respect will be discussed later. Table V gives values of the I.H.P. per sq. in. of bore^2 for various values of the piston speed and M.I.P. commonly adapted. The I.H.P. of a proposed cylinder is found by multiplying the square of the bore by the appropriate coefficient from Table V in the case of a two stroke and by half that figure in the case of a four stroke cylinder.

TABLE V

PISTON SPEEDS—FEET PER MINUTE

M.I.P. lb./in.²	600	650	700	750	800	850	900	950	1000
80	0·572	0·620	0·667	0·715	0·762	0·810	0·857	0·906	0·954
85	0·607	0·658	0·708	0·760	0·810	0·860	0·912	0·962	1·01
90	0·643	0·697	0·750	0·805	0·858	0·912	0·966	1·02	1·07
95	0·680	0·736	0·794	0·850	0·907	0·964	1·02	1·08	1·13
100	0·715	0·775	0·834	0·894	0·953	1·01	1·07	1·13	1·19
105	0·751	0·814	0·876	0·940	1·00	1·06	1·13	1·19	1·25
110	0·786	0·852	0·917	0·983	1·05	1·11	1·18	1·25	1·31

The tabulated figures are values of I.H.P. per in.² of cylinder bore² for two stroke engines. For four stroke engines divide the values by 2.

Example: The I.H.P. of a 30 in. four stroke cylinder working with a M.I.P. of 80 per sq. in. and piston speed of 900 ft. per minute is :—

$$30^2 \times 0{\cdot}857 \div 2 = 386 \text{ I.H.P.}$$

Piston Speeds.—For trunk piston land engines of the open type the piston speed usually varies from 600 ft. per minute with very small engines of about 8 in. bore to 900 ft. per minute with large engines of about 23 in. bore. A linear relation between bore and piston speed between these limits is given by the formula

$$\text{Piston speed} = 460 + 17{\cdot}6 \text{ (bore)},$$

which represents good average practice.

With high speed forced lubricated trunk engines the practices of the different makers are not so consistent but appear on the average to be based on an increase of about 20% above the figures quoted above for slow speed engines. Some makers adopt a uniform piston speed of about 800 ft. per minute for all sizes, but greater uniformity of reliability would appear to be obtained by a graduated scale of piston speeds, as indicated above.

Mean Indicated Pressure.—Assuming that in every case the engine is or may be required to run continuously at full load, then the M.I.P. at rated full load is mainly dependent on the cylinder bore in the case of four stroke engines for the following reasons :—

1. Effective cooling of the cylinder walls, covers and pistons becomes increasingly difficult as the size of the cylinder is increased on account of the increased length of stream lines through which the heat has to be conducted.
2. As the cylinder bore is increased it becomes more difficult to obtain an overload without smoke. With a 9 in. cylinder it is possible to obtain an M.I.P. of 140 lb. per sq. in. with an invisible exhaust and a rated M.I.P. of 105 lb. per sq. in. is permissible. With a 25 in. cylinder 120 lb. per sq. in. is about the limit, and it is found advisable to restrict the rated M.I.P. to about 87 lb. per sq. in. at nominal full load.

Table VI below gives the nominal full load M.I.P. for four stroke engines for continuous running and provides for occasional overloads of short duration, amounting to about 20%.

TABLE VI

Bore of Cylinder, ins.	8	10	12	14	16	18	20	25	30
M.I.P. at rated full load . .	110	107	105	102	99	96	94	87	80

Table VI applies more particularly to land engines. For large marine engines fitted with cooled pistons slightly higher figures are sometimes used.

With two stroke engines another factor enters into the question, viz., the efficiency of the scavenging process and the excess of air passed into the cylinder. With uncontrolled ports and assuming a liberal surplus of air the maximum M.I.P. obtainable is somewhere in the neighbourhood of 80% of that obtainable with a four stroke engine of the same size. Apart from the maximum pressure obtainable, a reduction of the M.I.P. by 15%, as compared with a four stroke engine, places both cycles on an approximate equality so far as the mean temperature is concerned. A reduction of the figures given in Table VI by 15% therefore appears rational and gives results which agree fairly closely with published information of conservative designs.

With supercharge devices, either in the form of valves in the cover or valve controlled ports in the liner, the mean pressure obtainable is only limited by the capacity of the scavenge pump and the limits imposed by the designer on the compression pressure. With such engines M.I.P. of 170 lb. per sq. in. have been obtained, and it is evident that with the supercharge system high mean pressures are not necessarily associated with high mean cycle temperatures, and no table can be laid down for the M.I.P. permissible in such cases, but the following formula is suggested :—

$$\text{M.I.P.} = P_1 \frac{L}{S} \cdot \frac{p}{p_0} \quad \text{———(2)}$$

where,

P_1 = Permissible M.I.P. for a four stroke engine of the same bore.

S = Stroke.

L = That amount of the stroke which remains to be performed after the scavenge ports or valves are closed.

p_0 = Atmospheric pressure (absolute).

p = Pressure (absolute) at the point of closing of the scavenge ports or valves (usually 1 to 2 lb. in excess of the pressure of the scavenge air).

Example :

Bore = 20 in.

L = 0·75 S.

p = 19 lb. per sq. in. abs.

From Table VI, $P^1 = 94$ lb. per sq. in.

Therefore M.I.P. $\frac{94 \times 0 \cdot 75 \times 19}{14 \cdot 7} = 91$ lb. per sq. in.

Both the above rules for two stroke engines assume that the design of the scavenging apparatus is consistent with the efficient expulsion of the exhaust gases.

Literature. — "Diesel Engine Cylinder Dimensions." — *Engineering*, September 26th, 1913.

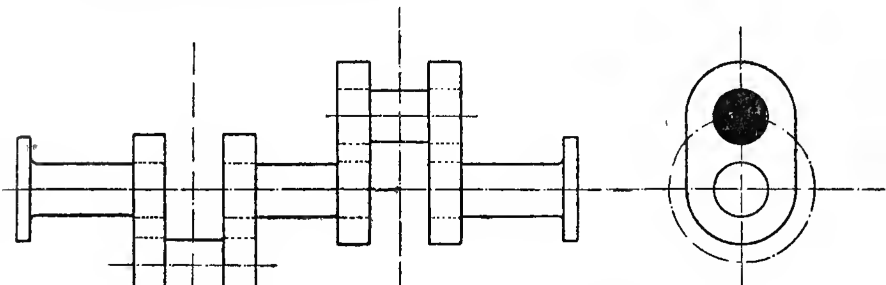
Richardson, J., "The Development of High Power Marine Diesel Engines."—Junior I. E., April 20th, 1914. This paper contains a great deal of information regarding the dimensions, weights, and capacities of the leading types of Diesel Engines.

CHAPTER V

CRANK-SHAFTS

Material.—For Marine Diesel Engine crank-shafts the usual material is open hearth steel having a tenacity of 28 to 32 tons per sq. in. and minimum elongation of 25 to 29% on 2 in., the lower minimum elongation being associated with the higher tenacity.

For stationary engines it is more usual to employ steel of a tenacity of 34 tons and upwards, specifying a minimum elongation of 25% in 2 in.



Centres of Cranks .	2·55	Thickness of Webs .	0·41
Diameter of Shaft .	0·63	Width of Webs .	1·32
Length of Crank-pin	0·63		

FIG. 21.

General Construction.—The majority of Diesel Engine crank-shafts are turned from solid forgings. When the ratio of stroke to bore exceeds about 1·8 a built-up shaft is sometimes used, and typical proportions are given on Fig. 21, the unit being the bore of the cylinder.

Fig. 22 shews a solid forged crank-shaft for a two cylinder stationary engine.

For four stroke stationary engines the shaft is usually of one piece when the number of cylinders does not exceed four.

For six cylinder stationary engines the usual practice is to divide the shaft into two sections, arranging the cam-shaft drive in the centre. Occasionally the one piece arrangement is adopted with the cam-shaft drive at one end, preferably the fly-wheel end. This makes the neater arrangement, and as no difficulties appear to occur in practice, probably the only disadvantage is the expense of replacing the whole shaft in the event of failure. The turning of long crank-shafts offers no difficulties provided a modern crank-shaft lathe is available. For four stroke marine engines of six and eight cylinders the usual arrangement is two strictly interchangeable sections of shaft. With two stroke marine engines the cylinders are arranged in pairs, with a section of shaft to each pair, the cranks of which are placed at 180°. This arrangement complies

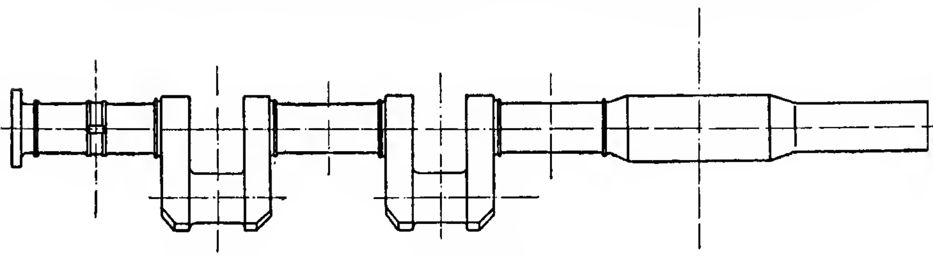


FIG. 22.

with the requirements of good balance and equal division of impulses, and the fact of both cranks of a section being in a plane facilitates both forging and machining. To secure interchangeability the number of bolts in each coupling should be either equal to or a multiple of the number of cylinders.

Arrangement of Cranks and Order of Firing.—

1. Two cylinder four stroke engines.

The cranks are commonly arranged on the same centre and the cylinders fire alternately at equal intervals, thus sacrificing balance to equal spacing of impulses. Arranging the cranks at 180° does not very materially affect the degree of uniformity and has the advantage of balancing the primary forces, but the primary couples are unbalanced. The latter would appear to be the lesser evil.

2. Three cylinder four stroke engines.

Cranks at 120° and firing periods follow at equal intervals of

240°. Primary and secondary forces are balanced, but unbalanced primary and secondary couples exist. The two latter do not appear to be very serious so far as their effects in producing vibrations are concerned.

3. Four cylinder four stroke engines.

The most common arrangement is to have all the cranks in one plane, the inner pair of cranks being on the same centre and the two outer cranks at 180° to them. Ignitions follow at intervals of 180°. The primary forces, and both the primary and secondary couples, are balanced, but the secondary forces are completely unbalanced. Vibration troubles are common with this arrangement. By arranging the cranks as for a two cycle engine (*q.v.*) both primary and secondary forces can be balanced at the expense of unequal spacing of ignitions.

4. Six cylinder four stroke engines.

In principle each half of the shaft represents the optical image of the other half as seen in a mirror placed at the centre of the engine in a plane at right angles to the centre line of the shaft, the cranks in each half being at 120° to each other, as for a three cylinder engine. The primary and secondary couples generated by each half of the engine mutually cancel each other, so that complete balance is obtained so far as primary and secondary forces and couples are concerned.

5. Eight cylinder four stroke engines.

The same principle of equal but opposite handed shaft halves applies to this case also. Each half consists of four cranks, the outside pairs of which are at 180° and the planes containing these pairs being at right angles. (See Fig. 23, which also shews an alternative arrangement.)

Each half of the engine is balanced for forces and the two halves balance each other for couples. Similar arrangements are possible for higher even numbers of cylinders.

6. Four cylinder two stroke engines.

The arrangement for cranks is the same as that described for one half of the shaft for an eight cylinder four stroke engine.

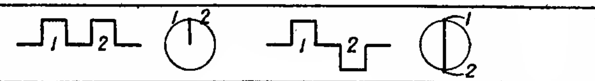
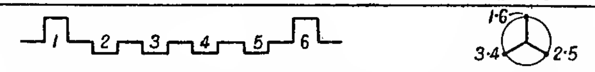
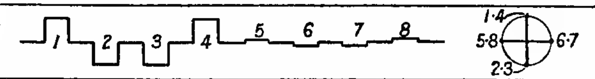
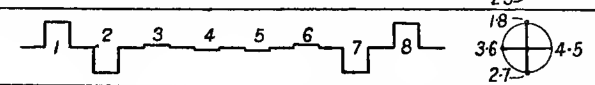
7. Six cylinder two stroke engines.

The arrangement consists of three pairs of cranks, the individuals of each pair being at 180° and the planes containing the pairs being at 120° to each other. Secondary couples only are out of balance.

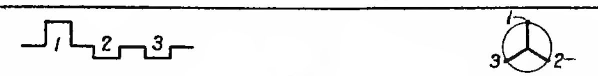
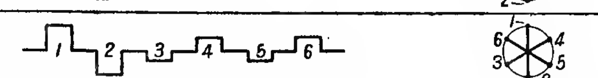
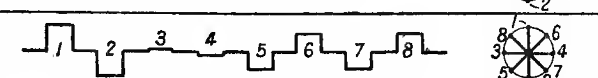
8. Eight cylinder two stroke engines.

Similar to six cylinder engine, but planes containing pairs of cranks at angles of 45°. Primary and secondary couples out of balance. With two stroke engines, owing to the fact that no two cranks are on the same centre, the order of firing is determined by the angular position of the cranks only. With four stroke engines, on the other hand, having four, six or eight cylinders for every firing point there is a choice of two cylinders and the orders of firing generally adopted are based on the principle of placing consecutively firing cylinders as remote as

4 Stroke Engines.

No. of Cylinders	Arrangement of Cranks.	Order of Firing
2		12 12
3		132
4		1243
6		153624
8		16284735
8		15268473

2 Stroke Engines.

3		123
4		1423
6		145236
8		16472538

Figs. 23, 24.

possible, so as to avoid local accumulation of elastic strain due to the reaction at the bearings. The usual arrangements and sequences of cranks, and also the order of firing, are shewn diagrammatically on Figs. 23 and 24 for four stroke and two stroke engines respectively.

Lubrication.—The question of the lubrication of Diesel Engines is partly influenced by the adoption of crossheads and guides distinct from the piston. So long as the trunk piston is used forced lubrication is at a disadvantage, on account of the difficulty which is experienced in preventing the splashing of oil on to the cylinder walls and disadvantages arising from the mixing of carbonised oil from the cylinder with that used for the bearings. The use of a crosshead, on the other hand, enables the cylinder to be isolated from the crank chamber, and forced lubrication can then be used with the same success which attends its application to high speed steam engines. To obtain the maximum benefit from this system it is necessary for the crank-case to be carefully designed to prevent loss of oil, either in the form of splash or impalpable mist, and also to prevent the ingress of grit. It is a mistake to suppose that a copious supply of lubricant under pressure necessarily eliminates wear. Only by the absolute exclusion of grit from the crank-case and from the entire lubricating system can wear be reduced to a minimum. For this reason it would seem desirable for all lubricating oil pumps, filters, etc., to be located within the crank-case itself. An exception to this rule may advantageously be made in the case of oil coolers if these are fitted. The presence of cold bodies within the crank-case results in the condensation of water, which mixes with the oil unless special arrangements are made to cope with this difficulty.

So far as economy of lubricating oil is concerned, ordinary ring lubrication for the main bearings and the centrifugal banjo arrangement for the big ends leave nothing to be desired. With suitable arrangements for filtering the oil which is drained from the crank-pit, and using over and over again, the nett lubricating oil consumption is readily kept below 0·002 lb. per B.H.P. hour (trunk piston engines). When forced lubricating is applied to high speed trunk engines the consumption is frequently higher. Efficient use of non-forced lubrication necessitates certain special features in connection with the crank-shaft. Rings are turned on the latter at each end of

each journal to throw the squeezed out oil into suitable catcher grooves in the bearing brasses, whereby it is returned to the oil well instead of being thrown off by the crank webs. As these oil throwers have been shewn in some cases to weaken the shaft at its already weakest point, they should be designed so as not to interfere with a good radius between the journal and the web. Fig. 25 shews a section through such an oil thrower.

Fig. 26 shews a crank fitted with centrifugal banjo lubricator. The oil hole leading to the surface of the crank-pin is preferably

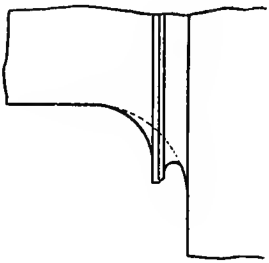


Fig. 25.

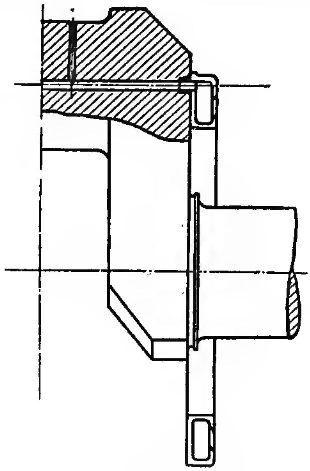


Fig. 26.

drilled at an angle of about 30° in advance of the dead centre, so that the upper connecting rod brass receives a supply of oil just before the ignition stroke.

With forced lubrication oil throwers and catchers are not usually fitted, but the shaft requires to be drilled to conduct oil from the journals to the crank-pins. Two systems of drilling are shewn in Figs. 27 and 28.

Details.—(1) **Webs.**—Various types of solid forged webs are shewn in Figs. 29, 30 and 31. The two ends of the straight-sided webs are sometimes turned from the journal and crank-pin centres respectively. Weight can be reduced slightly by turning the two ends at one setting from a centre midway between these two points. Triangular-shaped segments are

usually turned off the projecting corners of the webs, and this reduced weight facilitates feeling the big ends of the connecting rods and gives more clearance for the indicating gear. Balance

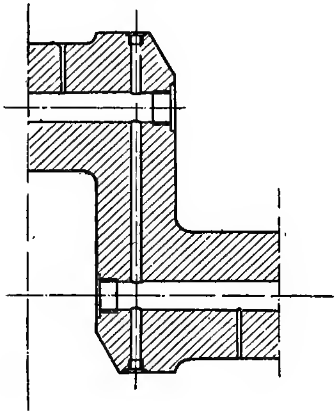


FIG. 27.

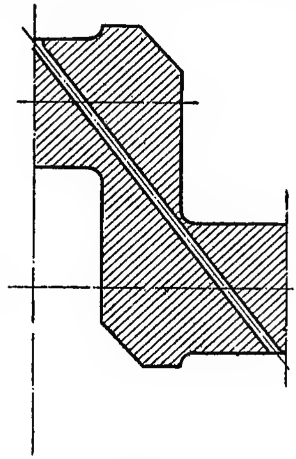


FIG. 28.

weights are frequently fitted to one and two cylinder engines to balance the revolving weight of the crank-pins, the big ends, and the otherwise unbalanced portion of the crank webs. The chief difficulty in designing a balance weight is usually to get a

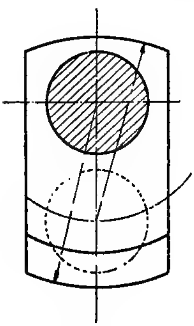


FIG. 29.

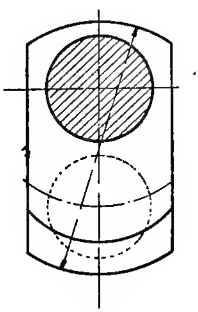


FIG. 30.

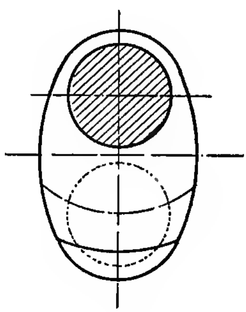


FIG. 31.

sufficiently heavy weight in the space available. The magnitude of the balance weight required is equal to the weight to be balanced (i.e. weight of crank-pin plus about 0·65 of the total weight of the connecting rod and about half the weight of the

webs) multiplied by the radius of the crank and divided by the radius measured from the centre of the shaft to the centre of gravity of the balance weight. The problem thus resolves itself into a matter of trial and error. Various modes of securing balance weights are illustrated in Figs. 32, 33 and 34. The bolts, or other form of attachment, should be sufficiently strong to carry the centrifugal force of the weight with a low stress.

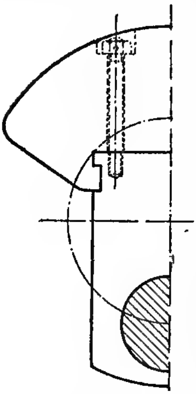


Fig. 32.

(2) **Couplings.**—The couplings connecting the sections of a shaft are made with spigot and faucet joints, the spigots being turned off after the bolts have been fitted. The bolts belonging to the coupling to which a gear-wheel is fitted are usually made of additional length, and used to secure the wheel. If separate means of securing the wheel are used the interchangeability of the sections of shaft is prejudiced. This is of small importance where land engines are concerned. With marine engines, where it is desirable to carry a spare section of shaft, the

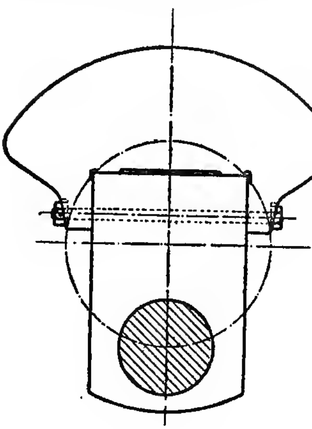


Fig. 33.

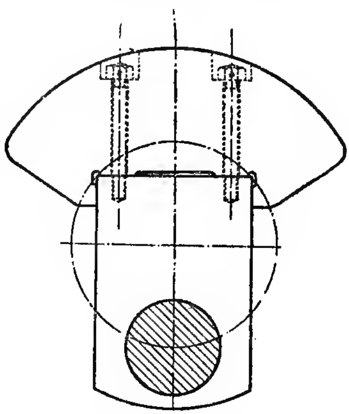


Fig. 34.

latter should be provided with any keyways, extra bolt-holes, etc., requisite to enable it to replace any section of the shaft in the event of failure, with a minimum of fitting.

When heavy fly-wheels are fitted, as for instance with dynamo

drives, an outer bearing is sometimes placed between the fly-wheel and the driven shaft. The coupling used to connect the projecting end of the crank-shaft to the drive may conveniently be of the common cast iron flanged type, provided with a shrouding to cover the nuts and bolt-heads. See Fig. 35.

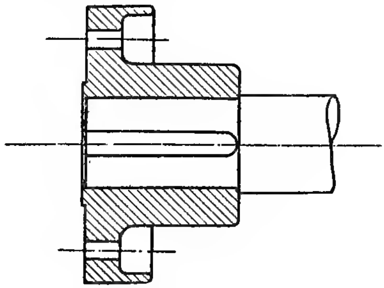


FIG. 35.

Occasionally this outer coupling is forged with the shaft, and if precautions are taken to ensure that the coupling is free of all bending moment it may be proportioned to the twisting moment only. This arrangement is a convenient one when the cam-shaft drive is at the fly-wheel end of the engine, as the small diameter of the coupling enables the crank-shaft gear-wheel to be passed over the coupling in one piece. See Fig. 36.

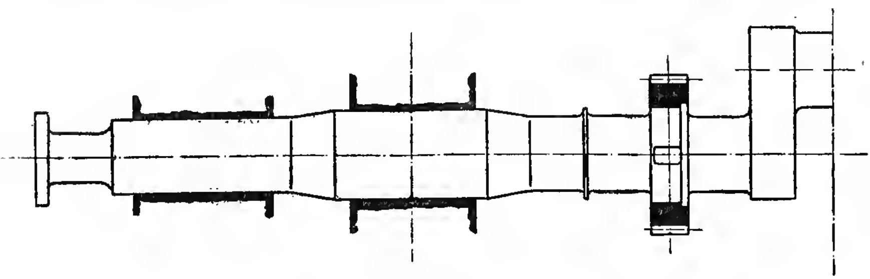


FIG. 36.

(3) **Air Compressor Cranks.**—When the air compressor is supplied by another manufacturer the compressor crank is usually designed by the latter. This arrangement is not always quite satisfactory as the apparatus by means of which the compressor is driven at the shop test may be more favourable for satisfactory running than the arrangements which are

made for the reception of the compressor on the engine. In any case close co-operation should exist between the two manufacturers to produce a satisfactory job between them. The main points to be insisted on are rigidity and truth of alignment. Figs. 37 and 38 illustrate two satisfactory designs of crank.

(4) **Scavenger Cranks.**—When the air compressor is driven by an overhung crank extending from the scavenger crank-shaft, the latter must be made far stiffer than would be required from considerations of strength alone, in order to keep the deflection of the overhung crank within small limits. In this case it is not unusual to make the scavenger crank the same diameter as the main crank-shaft. When two scavengers are

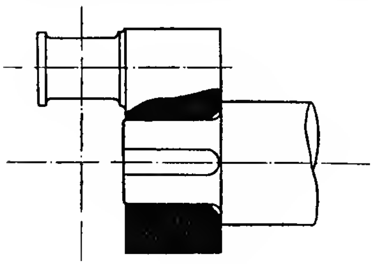


Fig. 37.

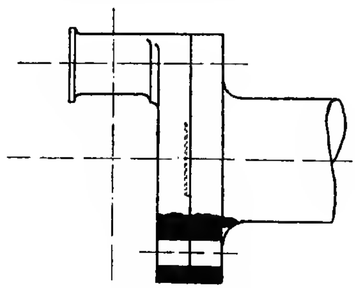


Fig. 38.

provided the cranks are placed at 90° to equalise the discharge of air.

Proportions.—In view of the fact that the straining actions causing failure of crank-shafts are mainly due to bending moments caused by unequal level of the bearings, the most consistent results are obtained when discussing the proportions found in practice by expressing all dimensions in terms of the cylinder bore. As regards marine crank-shafts, the designer has little latitude, as minimum dimensions are fixed by the rules of insurance societies. To evaluate these rules the distance between centres of cylinders must be determined, and this figure is usually about twice the cylinder bore in the case of four stroke engines and about 2 to 2·4 times the bore in the case of two stroke engines. The diameter of the shaft usually works out at about 0·62 for four stroke and 0·65 for two stroke engines. These approximate figures are merely quoted here for comparison with those relating to land engines.

Proportions of Four Stroke Land Engine Crank-shafts.—Typical proportions for a slow speed engine are given below, the unit being the cylinder bore :—

Diameter of journal and crank-pin	0·53
Length of journal	1·15
Length of crank-pin	0·53
Width of web	0·80
Thickness of web	0·28

Proportions of an exceptionally strong shaft are given below :—

Diameter of crank-pin and journal	0·57
Length of journal	0·88
Length of crank-pin	0·70
Width of web	0·92
Thickness of web	0·32

Mr. P. H. Smith, in a paper read by him before the Diesel Engine Users' Association, July, 1916, recommends the following proportions applying to 34-ton steel on the understanding that certain precautions are taken to keep the bearings in line.

Diameter of crank-pin and journal	0·525 to 0·54
Length of journal	0·75 ,, 0·80
Length of crank-pin	0·524 ,, 0·54
Thickness of web	0·32 minimum

Mr. Smith points out that Diesel Engine crank-shafts almost invariably fail at the webs, and the thickness he proposes for the latter is the greatest the author has found in practice.

According to Mr. Smith's minimum figure for the shaft diameter and web thickness, and taking the width of the web to be 0·8 of the cylinder bore, the relative strengths of the journal and the web to resist bending are as

$$\frac{\pi \times 0{\cdot}525^3}{32} \text{ to } \frac{0{\cdot}8 \times 0{\cdot}32^2}{6}$$

$$= 1 : 0{\cdot}96,$$

so that a shaft based on these proportions will be of nearly equal strength throughout if the effects of radii and changes of shape are neglected. Crank-shafts for two stroke land engines are generally of slightly larger diameter than those for four stroke engines. Different examples give figures varying from 0·55 to 0·59 of the cylinder bore. Great difference of opinion exists as to the size of the radii between journals and

crank-pins and crank webs. A radius of 0·07 of the shaft diameter is good average practice, though some designers prefer 0·15 and others are satisfied with 0·04.

Calculation of Stresses in the Crank-shaft.—It is as well to state at the outset that the problem of determining the stresses in a multi-crank-shaft is rather laborious, if done conscientiously and actual designs are more often than not based on experience pure and simple, without reference to comparative calculations other than of simple proportion. Supposing a suitable analysis to have been made for a correctly aligned shaft, the whole calculation would require to be revised before the results could be applied with accuracy to the case of a shaft of which the bearings were at different heights owing to the unequal wear of the white metal or flexure of the bedplate. This latter consideration is of itself valuable in emphasising the need of massive foundations where land engines are in question and the desirability of providing an extremely rigid framework in marine designs. Probably the severest condition with which a Marine Diesel Engine has to contend is the deflection of the hull due to variations of cargo loading, etc. A stiff seating in way of the engine is doubtless helpful, but the surest way of avoiding deflection troubles is to design the engine framework in the form of a deep box girder. Apart from other considerations, this enables the engine to be erected in the shops once and for all, without the necessity of re-bedding the shaft after installation in the ship. The component parts of the crank-shaft, viz., journals, crank-pins and webs, are subject to bending and twisting actions which vary periodically as the shaft revolves. In the past it has been customary to compute equivalent bending or twisting moments corresponding to the calculated co-existing bending and twisting moments and to proportion the shaft accordingly. Recent experiments by Guest and others indicate that steel under the influence of combined bending and twisting begins to fail when the shearing stresses, as calculated from the formula quoted below, attain a definite value (about 12,000 lb. per sq. in. for very mild steel under alternating stress) which is independent of the relative amounts of bending and twisting.

$$\text{Maximum shear stress} = \tfrac{1}{2}\sqrt{4 f_s^2 + f_n^2}$$

Where f_n = Normal stress due to bending.

f_s = Shear stress due to twisting.

The equivalent twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of the actual bending and twisting moments is given by

$$T_E = \sqrt{T^2 + B^2}$$

Where T_E = Equivalent twisting moment.
T = Actual twisting moment.
B = Bending moment.

A good approximation to the twisting moment at any point of the shaft at any degree of revolution is obtained by combining in correct sequence the twisting moment curves corresponding to all cylinders "for'd" of the section under consideration. In future the terms "forward" and "aft" will be used to denote the compressor and fly-wheel end of the engine respectively regardless of whether the engine under consideration is of marine or land type. The negative twisting moment due to mechanical friction of the moving parts is almost always neglected. That due to the compressor is sometimes allowed for. When dealing with the stress in a crank-pin it should be borne in mind that the twisting moment due to any cylinder is not transmitted through its own crank-pin. For example, if the cranks are numbered as usual from the compressor end, the twisting moment in No. 3 crank-pin is that due to cylinders Nos. 1 and 2. The calculation of bending moments is by no means straightforward, and the methods adopted form the distinguishing features of the systems of crank-shaft calculation described below.

Fixed Journal Method of Crank-shaft Calculation.—The assumption underlying this method is that each journal is rigidly fixed at its centre and that the section of shaft between two journals may therefore be treated as a beam encastré at its ends. The assumption of fixed journals would be true for a row of cylinders all firing at the same time. For ordinary conditions, however, the assumption would only hold good if the bearings had no running clearance and were capable of exerting a bending effect on the shaft by virtue of their rigidity. Apart from the fact that the construction of bearings and bearing caps does not suit them for this heavy duty, examination of the bearing surface of well-worn bearings reveals no trace of such cornering action and justifies the view that the bearings merely fulfil their proper functions of carrying thrust in one direction at a time.

Free Journal Method.—With this system each crank is supposed to be loaded at the centre of the crank-pin and supported freely at the centre of the journals, so that the maximum bending moment occurs at the centre of the crank-pin. This assumption would be approximately true for a single cylinder engine if the weight of the fly-wheel and the influence of outboard bearings are neglected. With this method the twisting moment is of very secondary importance, in many cases almost negligible. Comparing this system with that described above, it will be seen that both involve the construction of twisting moment diagrams, though the accuracy of the result is of less importance in the case of the free journal method.

In the following articles a four throw crank-shaft will be investigated on somewhat different lines, with a view to eliminating as many unjustifiable assumptions as possible.

Stress Calculation for a Four Throw Crank-shaft

Data:

Type of engine	four stroke
Number of cylinders	four
Bore of cylinders	10 in.
Stroke	15 in.
Revolutions per minute	300
Connecting rod	5 cranks long
Maximum pressure at firing dead centre,	500 lb. per sq. in.
Diameter of journals and crank-pins	5·25 in.
Length of journals	8 in.
Length of crank-pins	5·5 in.
Thickness of webs	3·25 in.
Width of webs	9 in.
Centres of cylinders	8+5·5+6·5=20 in.
Weight of piston	170 lb.
Weight of connecting rod	190 lb.
Weight of crank-pin	33 lb.
Weight of unbalanced parts of two crank-webs	110 lb.

The method employed in the following investigation is to calculate the values of the forces acting on the shaft when one crank is at its firing top dead centre. The reactions on the bearings will be calculated on the assumption that the centres of the journals remain level and all loads will be treated as concentrated. The effect of unequal level of bearings will also

be investigated. In computing the forces acting on the crank-shaft the dead weight of the latter, and also that of the running gear, fly-wheel, etc., will be neglected and the effect of the air compressor will not be considered, nor will the small exhaust pressure remaining in the cylinder which completes its exhaust stroke at the same instant that the cylinder under consideration begins its firing stroke, so that the forces to be dealt with are :—

(1) Those due to cylinder pressure.
(2) Centrifugal force of revolving parts.
(3) Inertia force of reciprocating parts.

These will now be calculated :—

Weight of revolving part of connecting rod, $0{\cdot}65 \times 190$	124 lb.
Weight of unbalanced part of crank webs .	110 lb.
Weight of crank-pin	33 lb.
Total weight of unbalanced revolving parts	267 lb.

Weight of reciprocating part of connecting rod, $0{\cdot}35 \times 190$	66 lb.
Weight of piston	170 lb.
Total weight of reciprocating parts .	236 lb.

Centrifugal acceleration in the crank circle is :—

$$w^2r = \left(\frac{2\pi.300}{60}\right)^2 \times \frac{7{\cdot}5}{12} = 615 \text{ ft. per sec.}^2$$

Therefore centrifugal effect of revolving parts

$$\frac{267 \times 615}{g} = 5100 \text{ lb.}$$

Inertia effect of reciprocating parts at top dead centre :—

$$\frac{236 \times 615}{g} \times 1\tfrac{1}{5} = 5400 \text{ lb.}$$

Inertia effect of reciprocating parts at bottom dead centre :—

$$\frac{236 \times 615}{g} \times (1 - \tfrac{1}{5}) = 3600 \text{ lb.}$$

Combined centrifugal and inertia effect at top dead centre

$$= 5100 + 5400 = 10{,}500 \text{ lb. upwards.}$$

Combined centrifugal and inertia effect at bottom dead centre

$$= 5100 + 3600 = 8700 \text{ lb. downwards.}$$

Maximum load due to cylinder pressure

$$=0{\cdot}785\times10^2\times500=39{,}000\text{ lb.}$$

Resultant downward effect of pressure, centrifugal force and inertia force at firing dead centre

$$=39{,}000-10{,}500=28{,}500\text{ lb.}$$

Method of Calculating the Reactions at the Bearings.—Let A 1 (Fig. 39) represent the centre of the crank-shaft, A C E G I being the centres of the journals and B D F H those of the crank-pins.

$R_2\,R_4\,R_6\,R_8$ are the applied forces due to cylinders 1, 2, 3, 4 respectively.

$R_1\,R_3\,R_5\,R_7\,R_9$ are the reactions at the bearings unknown in magnitude and direction. Under the influence of these forces the centre line of the shaft assumes some deflected shape,

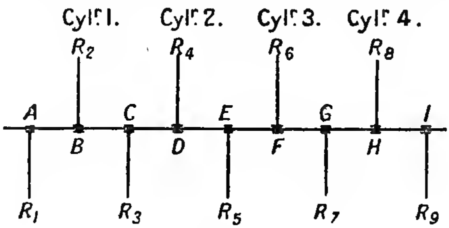


FIG. 39.

and the deflection at any point above or below the straight line joining A 1 is equal to the sum of the deflections at the same point which would be produced by each of the forces $R_2\,R_3\,R_4\,R_5\,R_6\,R_7\,R_8$ acting alone, supposing the shaft supported freely at A and 1. At C E and G the sum of these deflections must be zero if the bearings are level (ignoring the effect of running clearance). Let $c_2\,e_2$ and g_2 be the deflections at C E and G due to unit load applied at B (the position of R_2), assuming the shaft supported freely at A and 1.

$$\left.\begin{matrix} c_3\,e_3 \text{ and } g_3 \\ c_4\,e_4 \text{ and } g_4 \\ c_5\,e_5 \text{ and } g_5 \\ \text{etc., etc.} \\ c_8\,e_8 \text{ and } g_8 \end{matrix}\right\} \begin{matrix}\text{are the corresponding deflections at C E and G} \\ \text{due to unit load applied at C D E, etc.}\end{matrix}$$

The values of $c_2\,e_2$ and g_2, etc., are readily found by the usual formulæ for the deflection of beams.

Now since the total deflection at C E and G is zero, the following equations hold good :—

$$R_2 c_2 + R_3 c_3 + R_4 c_4 + R_5 c_5 + R_6 c_6 + R_7 c_7 + R_8 c_8 = 0 \text{——(1)}$$
$$R_2 e_2 + R_3 e_3 + R_4 e_4 + R_5 e_5 + R_6 e_6 + R_7 e_7 + R_8 e_8 = 0 \text{——(2)}$$
$$R_2 g_2 + R_3 g_3 + R_4 g_4 + R_5 g_5 + R_6 g_6 + R_7 g_7 + R_8 g_8 = 0 \text{——(3)}$$

In these three equations the only unknown quantities are $R_3\, R_5\, R_7$, which can therefore be determined. The remaining unknown reactions R_1 and R_9 are found by equating moments about 1 and A respectively. In solving the above equations downward forces and deflections will be considered positive and upward forces and deflections negative.

Determination of $c_2\, e_2\, g_2$, etc.—In determining these constants it will be assumed that the shaft deflects under load as though it were a cylindrical beam of the same diameter as the crank-pins and journals. Considering that the webs of a Diesel Engine crank are short, the presumption is probably not inaccurate, but it would be interesting to see this point investigated, as it could readily be, by means of models. So long as bearings at constant level are assumed, the actual value of the deflection is of no importance, as the method of calculation depends only on the relative deflection at the different points considered. The problem therefore resolves itself into finding the deflected form of a beam freely supported at each end under the influence of a concentrated load placed anywhere between the supports. This may be done by treating each end of the beam as a cantilever. The deflection of the end of a cantilever carrying a load at the end is given by :—

$$\text{Deflection at the end of cantilever in in.} = \frac{W.l^3}{3\,EI}$$

Where W = Load in lb.

E = 30,000,000 lb. per sq. in. (for steel).

I = Moment of inertia (transverse) of the section of beam in.

For the shaft under consideration :—

$$I = \frac{\pi}{64} \times 5{\cdot}25^4 = 37{\cdot}3 \text{ in.}^4$$

Fig. 40 shews the values of the deflection at various fractional points in the length of the cantilever, the deflection at the end being unity. These results are applied to the case of a beam as follows :—

Let AB be a beam (Fig. 41) supported at A and B and

carrying a load W_1 at any point C. The reaction at A is equal to $W_1 \frac{CB}{AB}$. Let this be denoted by W_2. At A erect a perpendicular AD equal to some convenient scale to the deflection of the cantilever AC, due to the load W_2 at its end. Draw the deflected shape of this cantilever (DC) by means of the proportions given

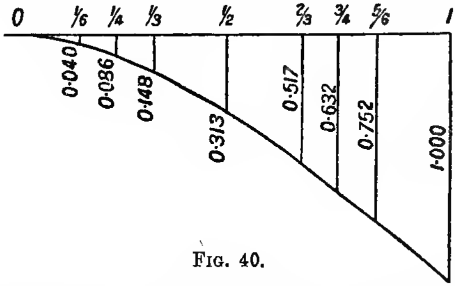


Fig. 40.

in Fig. 40. Proceed similarly with cantilever C B, obtaining the deflected shape C F. Join D and F, then the deflection of the beam at any point X is the vertical intercept " x " shewn in the figure.

Application to the Case in Hand.—The constants $c_2 \, e_2 \, g_2$ being the deflections at various points on a beam due to unit load applied at other various points, are independent of the system of loads and will therefore be dealt with before special

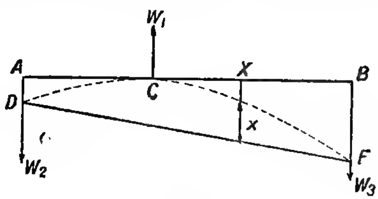


Fig. 41.

cases of loading are considered. In order to obtain manageable figures, the deflection will be reckoned in thousandths of an inch and the unit load will be taken as 10,000 lb. Deflection of cantilevered portion of shaft in thousandths of an inch is given by :—

$$\frac{W.(\frac{l}{10})^3}{3 \times 30 \times 37{\cdot}3} = \frac{W.(\frac{l}{10})^3}{3360}$$

Where W = Load in lb.

l = Length in in.

The diagram Fig. 42 shews the process of determining $c_2\, e_2\, g_2$, etc., in particular :—

Unit load at B = 10,000 lb., AB = 10″, BI = 70″

$$\text{Reaction at A} = \frac{10{,}000 \times 70}{80} = 8750 \text{ lb.}$$

$$\text{,, \quad ,, \quad I} = \frac{10{,}000 \times 10}{80} = 1250 \text{ lb.}$$

$$\text{Deflection of cantilever AB} = \frac{8750 \times 1^3}{3360} = 2{\cdot}6 \tfrac{\text{in.}}{1000}$$

$$\text{,, \quad ,, \quad IB} = \frac{1250 \times 7^3}{3360} = 127 \tfrac{\text{in.}}{1000}$$

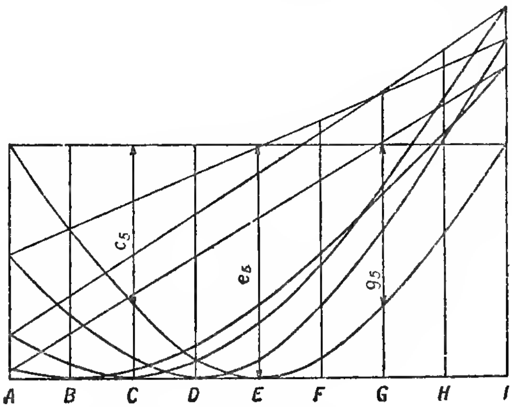


FIG. 42.

Deflected shapes of cantilevers drawn by plotting ordinates from the proportion given in Fig. 40.

By scaling the diagram :—

$$c_2 = 30{\cdot}5 \quad e_2 = 35 \quad g_2 = 20{\cdot}5$$

Since the deflection at G, due to a load at H, is the same as that at C, due to the same load at B, and so on, therefore :—

$$g_8 = 30{\cdot}5 \quad e_8 = 35 \quad c_8 = 20{\cdot}5$$

By the same methods (see Fig. 42) :—

$$c_3 = 51 \quad e_3 = 62 \quad g_3 = 39$$
$$g_7 = 51 \quad e_7 = 62 \quad c_7 = 39$$

$$c_4 = 65 \quad e_4 = 88 \quad g_4 = 55$$

$$g_6 = 65 \quad e_6 = 88 \quad c_6 = 55$$
$$c_5 = 65 \quad e_5 = 95 \quad g_5 = 65$$

These values could be computed with rather less trouble and with greater accuracy by the formulæ for the deflection at any point in a beam due to a load at any other point (see Morley's "Strength of Materials"). In more complicated cases, with a larger number of unknown reactions, the constants c, e, g require to be ascertained very accurately, otherwise large errors arise.

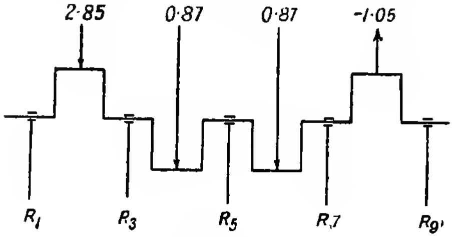


FIG. 43.

The conditions of loading will now be considered.

Case I. Crank 1 on Firing Dead Centre.—The magnitudes and directions of the applied forces are shewn in Fig. 43. Hence :—

$$R_2\,c_2=2{\cdot}85\times30{\cdot}5=87{\cdot}0 \qquad R_4\,c_4=0{\cdot}87\times65=56{\cdot}5$$
$$R_2\,e_2=2{\cdot}85\times35\quad=99{\cdot}5 \qquad R_4\,e_4=0{\cdot}87\times88=76{\cdot}5$$
$$R_2\,g_2=2{\cdot}85\times20{\cdot}5=58{\cdot}5 \qquad R_4\,g_4=0{\cdot}87\times55=48{\cdot}0$$

$$R_6\,c_6=0{\cdot}87\times55=48{\cdot}0 \qquad R_8\,c_8=-1{\cdot}05\times20{\cdot}5=-21{\cdot}5$$
$$R_6\,e_6=0{\cdot}87\times88=76{\cdot}5 \qquad R_8\,e_8=-1{\cdot}05\times35\quad=-37{\cdot}0$$
$$R_6\,g_6=0{\cdot}87\times65=56{\cdot}5 \qquad R_8\,g_8=-1{\cdot}05\times30{\cdot}5=-32{\cdot}0$$

From which

$$R_2\,c_2+R_4\,c_4+R_6\,c_6+R_8\,c_8=170{\cdot}0$$
$$R_2\,e_2+R_4\,e_4+R_6\,e_6+R_8\,e_8=215{\cdot}5$$
$$R_2\,g_2+R_4\,g_4+R_6\,g_6+R_8\,g_8=131{\cdot}0$$

Substituting these values in equations (1), (2), and (3), we obtain :—

$$51\,R_3+65\,R_5+39\,R_7=-170{\cdot}0 \text{——} (4)$$
$$62\,R_3+95\,R_5+62\,R_7=-215{\cdot}5 \text{——} (5)$$
$$39\,R_3+65\,R_5+51\,R_7=-131{\cdot}0 \text{——} (6)$$

From which

$$R_3=-2{\cdot}295 \quad R_5=-1{\cdot}402 \quad R_7=-0{\cdot}965$$

Equating moments about A :—

$$8\,R_9+7\,R_8+6\,R_7+5\,R_8+4\,R_5+3\,R_4+2\,R_3+R_2=0$$

$$\therefore \quad 8\,R_9 = 7\times1{\cdot}05 - 6\times0{\cdot}965 - 5\times0{\cdot}87 + 4\times1{\cdot}402 - 3\times0{\cdot}87 + 2\times2{\cdot}295 - 2{\cdot}850$$

Whence $R_9 = 0{\cdot}118$.

Equating moments about I :—

$$8\,R_1 + 7\,R_2 + 6\,R_3 + 5\,R_4 + 4\,R_5 + 3\,R_6 + 2\,R_7 + R_6 = 0$$

$$\therefore \quad 8\,R_1 = -7\times2{\cdot}85 + 6\times2{\cdot}29 - 5\times0{\cdot}87 + 4\times1{\cdot}402 - 3\times0{\cdot}87 - 2\times0{\cdot}965 + 1{\cdot}05$$

Whence $R_1 = -1{\cdot}079$.

Knowing all the forces, the bending moment at A B C, etc., can now be tabulated thus :—

Point.	Moments.	B.M. in in. lb.
A	. . . .	0
B	$10{,}510\times10$	105,100
C	$10{,}510\times20 - 28{,}500\times10$	−74,800
D	$10{,}510\times30 - 28{,}500\times20 + 22{,}950\times10$ . .	−25,200
E	$10{,}510\times40 - 28{,}500\times30 + 22{,}950\times20 - 8700 \times 10$	−62,600
F	$-2440\times30 + 10{,}500\times20 - 9{,}650\times10$ . .	30,300
G	$-2440\times20 + 10{,}500\times10$	56,200
H	-2440×10.	−24,400
I	. . . .	0

Since the bending modulus of the shaft is

$$\frac{\pi}{32}\times5{\cdot}25^3 = 14{\cdot}2 \text{ in.}^3$$

Therefore,

Maximum stress due to bending (occurring in No. 1 crank-pin at top firing centre) is equal to :—

$$\frac{105{,}100}{14{\cdot}2} = 7{,}400 \text{ lb. per in.}^2$$

Bending stress, according to fixed journal method :—

$$\frac{W.L}{8\ Z} = \frac{28{,}500\times20}{8\times14{\cdot}2} = 5020 \text{ lb. sq. in.}$$

According to free journal method :—

$$\text{Stress} = \frac{28{,}500\times20}{4\times14{\cdot}2} = 10{,}040 \text{ lb. sq. in.}$$

Case II. No. 2 Crank on Firing Dead Centre.—The magnitudes and directions of the applied forces are shewn in Fig. 44.

$$R_2=0{\cdot}870 \quad R_4=2{\cdot}850 \quad R_6=-1{\cdot}05 \quad R_8=0{\cdot}870$$

$$R_2\,c_2=0{\cdot}87\times30{\cdot}5=26{\cdot}5 \quad R_4\,c_4=2{\cdot}85\times65{\cdot}0=185{\cdot}0$$
$$R_2\,e_2=0{\cdot}87\times35{\cdot}0=30{\cdot}5 \quad R_4\,e_4=2{\cdot}85\times88{\cdot}0=251{\cdot}0$$
$$R_2\,g_2=0{\cdot}87\times20{\cdot}5=18{\cdot}0 \quad R_4\,g_4=2{\cdot}85\times55{\cdot}0=156{\cdot}5$$

$$R_6\,c_6=-1{\cdot}05\times55{\cdot}0=-58{\cdot}0 \quad R_8\,c_8=0{\cdot}87\times20{\cdot}5=16{\cdot}0$$
$$R_6\,e_6=-1{\cdot}05\times88{\cdot}0=-92{\cdot}5 \quad R_8\,e_8=0{\cdot}87\times35{\cdot}0=30{\cdot}5$$
$$R_6\,g_6=-1{\cdot}05\times65{\cdot}0=-68{\cdot}0 \quad R_8\,g_8=0{\cdot}87\times30{\cdot}5=26{\cdot}5$$

From which

$$R_2\,c_2+R_4\,c_4+R_6\,c_6+R_8\,c_8=171{\cdot}5$$
$$R_2\,e_2+R_4\,e_4+R_6\,e_6+R_8\,e_8=219{\cdot}5$$
$$R_2\,g_2+R_4\,g_4+R_6\,g_6+R_8\,g_8=133$$

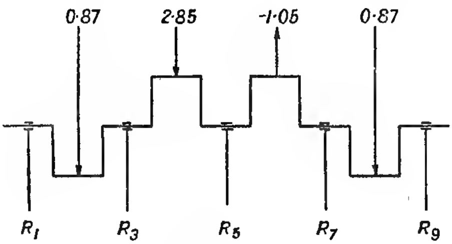


FIG. 44.

The three equations for determining $R_3\,R_5\,R_7$ are therefore :—

$$51\,R_3+65\,R_5+39\,R_7=-171{\cdot}5$$
$$62\,R_3+95\,R_5+62\,R_7=-219{\cdot}5$$
$$39\,R_3+65\,R_5+51\,R_7=-133$$

from which

$$R_3=-2{\cdot}005 \quad R_5=-1{\cdot}788 \quad R_7=1{\cdot}204$$

Taking moments about A and I,

$$R_1=-0{\cdot}161 \quad R_9=-0{\cdot}790$$

from which the following figures for the bending moments are obtained :—

Point.	Moments.	B.M. in in. lb.
A	. . . .	0
B	1610×10	16,100
C	$1610 \times 20 - 8700 \times 10$.	−54,800
D	$1610 \times 30 - 8700 \times 20 + 20{,}050 \times 10$ . . .	74,800
E	$1610 \times 40 - 8700 \times 30 + 20{,}050 \times 20 - 28{,}500 \times 10$	−80,600
F	$7900 \times 30 - 8700 \times 20 - 12{,}040 \times 10$ . . .	−57,400
G	$7900 \times 20 - 8700 \times 10$.	71,000
H	7900×10	79,000
I	. . . .	0

$$\text{Maximum bending stress } \frac{80{,}600}{14{\cdot}2} = 5670 \text{ lb. sq. in.}$$

Considerably less than in Case I.

Case III. Loading as in Case I, but bearings C and G supposed worn down 20 thousandths and bearing E 25 thousandths of an inch below the level of the line joining A I.

The difference in height of the bearings is very simply allowed for by putting 20, 25, and 20 respectively on the right-hand side of equations (1), (2), and (3) instead of zero.

The equations for determining $R_3\, R_5$ and R_7 then become :—

$$51\, R_3 + 65\, R_5 + 39\, R_7 = 20 - 170{\cdot}0 = -150{\cdot}0$$
$$62\, R_3 + 95\, R_5 + 62\, R_7 = 25 - 215{\cdot}5 = -190{\cdot}5$$
$$39\, R_3 + 65\, R_5 + 51\, R_7 = 22 - 131 \quad = -109$$

The following values are obtained for the reactions at the bearings :—

$$R_1 = -1{\cdot}412 \quad R_3 = -1{\cdot}610 \quad R_5 = -2{\cdot}135 \quad R_7 = -1{\cdot}820$$
$$R_9 = -0{\cdot}202$$

The bending moments are as follows :—

Point.	Moments.	B.M. in in. lb.
A	. . . .	0
B	$14{,}120 \times 10$	141,200
C	$14{,}120 \times 20 - 28{,}500 \times 10$	$-2{,}600$
D	$14{,}120 \times 30 - 28{,}500 \times 20 + 16{,}100 \times 10$ . .	146,000
E	$14{,}120 \times 40 - 28{,}500 \times 30 + 16{,}100 \times 20 - 8700 \times 10$	$-55{,}200$
F	$2020 \times 30 + 10{,}500 \times 20 - 18{,}200 \times 10$ . .	88,600
G	$2020 \times 20 + 10{,}500 \times 10$	145,400
H	2020×10	20,200
I	. . . .	0

Maximum bending stress $\frac{146{,}000}{14{\cdot}2} = 10{,}250$ lb. sq. in.

Comparing the above figures with those obtained in Case I, it will be seen that the maximum stresses have been increased to the extent of about 35% by the difference of level of the bearings. In view of the uncertainty which exists with regard to the deflection of cranked shafts the above figures are not strictly reliable, but the writer is of opinion that they under- rather than over-estimate the stresses.

Conclusions.—1. The value of the bending moments at a crank-pin on the top firing centre is greater for a crank-pin situate at one end of the shaft than that at one nearer the centre of the shaft.

2. The bending moments at certain crank-pins and journals may be as great or greater than the bending moment at the crank-pin which is receiving the greatest applied load.

3. No general rule for the bending moment at a Diesel Engine crank-pin or journal can represent the true state of affairs, but every different arrangement of cranks and number of cylinders requires to be investigated individually.

4. Difference of level of the bearings, due to wear or otherwise, gives rise to greatly increased bending moments, which can be calculated approximately in the manner described.

The methods of calculation which have been illustrated in

this chapter can be applied to cases involving any number of cranks, and the effects of fly-wheels, the rotors of electric generators, outboard bearings, etc., can be included. When the number of cylinders is three or less the weight of the fly-wheel is considerable, and cannot therefore be ignored. In these cases allowance should be also made for the practice of packing the outward bearing above the level of the engine main bearings. For engines of four cylinders and over, the weight of the fly-wheel and the presence of outboard bearings can probably be neglected with safety. An extended treatment of this subject will have to be reserved for a future occasion.

The labour involved in solving simultaneous linear equations increases as the square of the number of unknowns. For a description of a machine devised to do this work mechanically, see the "Treatise on Natural Philosophy," vol. i., Kelvin and Tait.

Graphical Determination of the Twisting Moments.—In the processes described below the following approximations have been made :—

1. The negative twisting moments due to the air compressor at the forward end of the engine have been neglected. These moments are small in comparison with the moments due to the working cylinders, and being opposite in direction to the maximum moments tend to reduce the latter by a small amount.

2. Moments due to the dead weight of the revolving and reciprocating parts have also been neglected. In a very large engine it would be advisable to take these into consideration, as other things being equal the dead weight of the running gear per square inch of piston area increases as the scale of the engine.

3. The twisting moments due to mechanical friction have been neglected, as (so far as the present writer is aware) the distribution and variation of the friction forces are not known with any exactitude, and in any case one is a little on the safe side in neglecting them. These friction moments of course accumulate as one passes from the forward to the aft end of the engine, where they amount in aggregate to about 15% of the mean indicated twisting moment, so their effect on the forward end of the shaft is quite negligible.

4. The moment of inertia of the fly-wheel has been assumed to be large in comparison with the fly-wheel effect of the revolving and reciprocating parts of the running gear. In cases where the fly-wheel is very small, or omitted altogether, as in some two stroke marine engines, the irregularities of turning effort are mainly absorbed by the angular acceleration of the crank masses. Allowance is readily made for this effect in the following manner. The combined twisting moment curve for all the cylinders is first found without allowance for fly-wheel effect, and a new zero line is taken at the height corresponding to the mean twisting moment. Ordinates measured to this new zero line represent fluctuations of the twisting moment from its mean value. These ordinates are now divided into segments proportional to the fly-wheel effects of the fly-wheel and crank masses. For example, if there are four cylinders and the moment of inertia of the fly-wheel is three times that of one set of crank masses, then the ordinates will be divided into seven parts, one part being applied in opposite sense to corresponding points on the twisting moment curve of each cylinder and the remaining three parts of each ordinate form the ordinates of a curve of the twisting moments absorbed by the angular inertia of the fly-wheel.

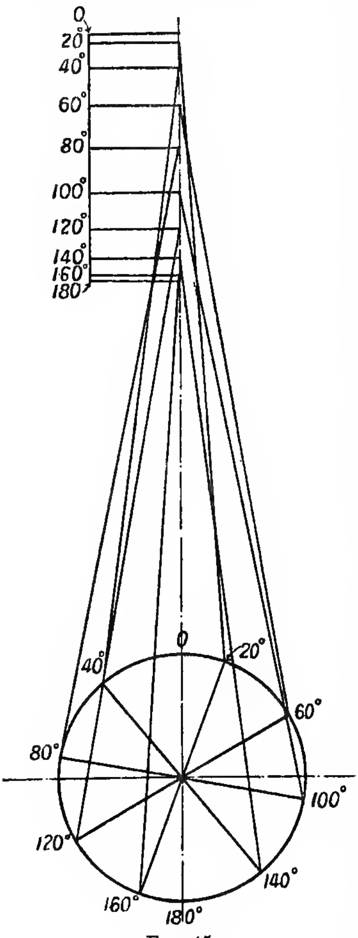


Fig. 45.

In the example worked out below it will be assumed that the fly-wheel is large compared with the crank masses, so that the process described briefly above is not necessary.

Fig. 45 is a skeleton diagram of the connecting rod positions for every 20 degrees of revolution of the crank-shaft for the determination of piston displacements.

Fig. 46 is a typical full load indicator card calibrated for pressures vertically and percentages of stroke horizontally. Points corresponding to each 20 degrees of revolution are marked on the diagram by scaling the piston displacements off Fig. 45.

On Fig. 47 cylinder pressures are plotted on a crank angle base from 0 to 720 degrees (four stroke engine). The pressure

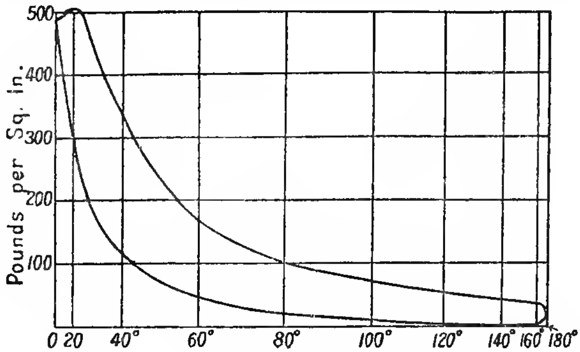


Fig. 46.

during the suction and exhaust strokes is assumed atmospheric. The inertia effect of the reciprocating parts per square inch of piston area is plotted from the following figures :—

Inertia effect at top dead centre $\dfrac{236 \times 615}{g}(1+\frac{1}{5})=5400$ lb.

" " bottom " $\dfrac{236 \times 615}{g}(1-\frac{1}{5})=3600$ "

" " $90°=\dfrac{236 \times 615}{g}\times\dfrac{1}{5}=$ 900 lb.

" " $45°=\dfrac{236 \times 615}{g}\times\dfrac{1}{\sqrt{2}}=3200$ lb.

Corresponding figures per sq. in. of piston area (78·5 sq. in.) are 69, 46, 11·5, and 41 lb. per sq. in. respectively.

The centrifugal forces of the revolving masses being radial produce no twisting effect on the shaft. The dotted line (Fig. 47) is the resultant of the pressure and inertia curves. The twisting moments are now computed as in the following table :—

Degrees from top dead centre.	Leverage in ins.	Resultant force in lb./in.^{2} of piston area.	Twisting moments in. lb.	Degrees from top dead centre.	Leverage in ins.	Resultant force in lb./in.^{2} of piston area.	Twisting moments in. lb.
0	0	418	0	360	0	−69	0
20	3·00	444	104,000	340	−3·00	−60	14,100
40	5·63	284	125,000	320	−5·63	−46	20,400
60	7·20	138	78,000	300	−7·20	−25	14,100
80	7·65	100	60,000	280	−7·65	0	0
100	7·13	90	50,000	260	−7·13	20	−11,200
120	5·85	90	41,400	240	−5·85	39	−17,900
140	4·08	88	28,200	220	−4·08	47	−15,100
160	2·10	85	14,000	200	−2·10	59	−9,750
180	0	73	0	180	0	73	0
360	0	−69	0	720	0	418	0
380	3·00	−60	−14,100	700	−3·00	232	−54,500
400	5·63	−46	−20,400	680	−5·63	69	−30,500
420	7·20	−25	−14,100	660	−7·20	22	−12,400
440	7·65	0	0	640	−7·65	22	−13,200
460	7·13	20	11,200	620	−7·13	32	−17,900
480	5·85	37	17,000	600	−5·85	42	−19,300
500	4·08	44	14,100	580	−4·08	44	−14,100
520	2·10	45	7,400	560	−2·10	45	−7,400
540	0	46	0	540	0	46	0

The leverage tabulated in the second column is found by the well-known graphical construction in Fig. 45, where the line of the connecting rod is produced (if necessary) to meet the horizontal line through the centre of the shaft, the intercept being the leverage required to the same scale as the rest of the diagram. The twisting moments are found by multiplying the leverage in inches by the resultant forces per sq. in. of

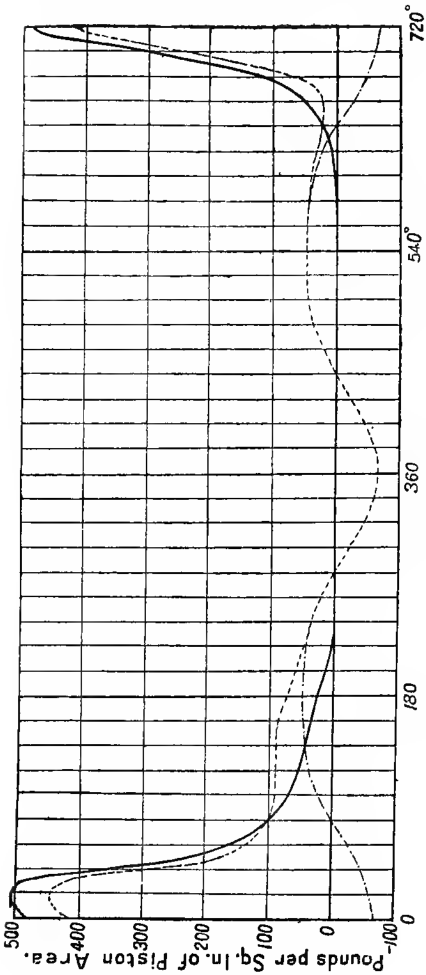


FIG. 47.

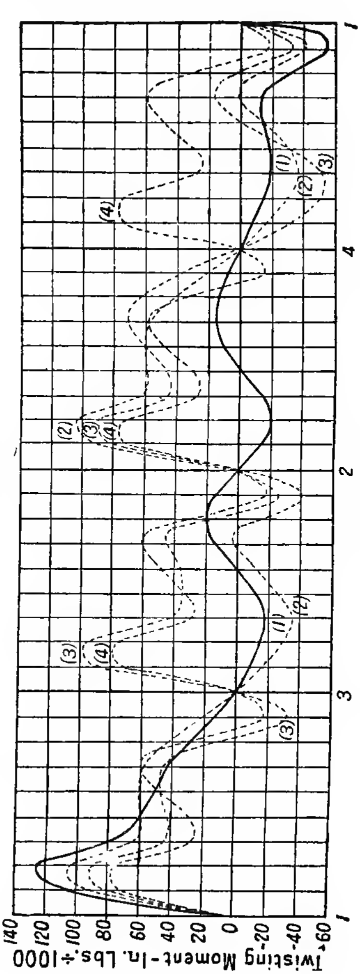


FIG. 48.

piston area in lb. per sq. in. and by the piston area in sq. in. (in this case 78·5 sq. in.).

Forces acting towards the crank are considered positive, whether they are expansion forces or otherwise, and those acting away from the crank negative. Assuming rotation clockwise, leverages to the right hand of the centre line are positive and those to the left negative. The signs of the moments then look after themselves according to the signs of their factors. It is not unusual to see the leverage in a case of this sort treated as though it were always positive. The disadvantage of this proceeding is that in order to get the signs of the moments correct those of the resultant forces have to be reversed at every dead centre, which besides being incorrect from a mathematical standpoint is inconvenient for the draughtsman and confusing to others. The tabulated values of the twisting moment are plotted in Fig. 48 (full line curve). Identical curves for cylinders 3, 4, and 2 could be plotted in their respective places at 180 degrees apart, in the order named, but are omitted for the sake of clearness. Dotted curve numbered 2 is the resultant of the curves belonging to cylinders 1 and 2. Dotted curve 3 is the resultant of the curves belonging to cylinders 1, 2, and 3, and so on. The simplest way of obtaining these resultants is to trace the primary curve on a piece of transparent paper and move it sideways into its required position for the next cylinder, and then for every required ordinate move the paper vertically (guided by the vertical degree lines) until the zero line coincides with the top of the ordinate of the curve to which it is required to add the effect of another cylinder. In this position prick through the top of the ordinate of the curve on the tracing-paper to the diagram underneath. These resultant curves enable the twisting moment at any crank-pin or journal at any angular position to be read off the diagram.

Combined Effect of Bending and Twisting.—It will be seen that the peaks of the twisting moment curves occur about 30 degrees after the dead centres, and that the results previously obtained for the bending moments with the cranks on dead centre apply very closely to this position also, so that the tabulated values of the bending moments at the various journals and crank-pins combined with the twisting moments existing at these points 30 degrees after the corresponding firing dead centres have been passed, represent the maximum

conditions of stress at the points in question. The conditions of bending when cranks 3 and 4 are on firing centre are of course the same as those obtaining when cranks 2 and 1 respectively are in that position, the order in which the bending moments occur being reversed.

For example, the bending moment at No. 2 crank-pin when No. 4 cylinder is firing is the same as the bending moment at No. 3 crank-pin when No. 1 cylinder is firing, and so on. A comparison of the following table with the twisting moment curves and the bending moments tabulated in the previous articles will make the matter clear.

The equivalent twisting moment equals $\sqrt{T^2+B^2}$, and is that twisting moment which would give the same shear stress as the maximum shear stress due to the combined action of twisting and bending moments actually obtaining.

Position.	Which crank 30° past firing dead centre.	Bending moment in. lb. (see previous tables).	Number of twisting moment curve.	Twisting moment in. lb. from curves.	Equivalent twisting moment in. lb.	Maximum shear stress lb. per sq. in.
A. Journal	—	—	—	—	—	—
B. Crank-pin No. 1	No. 1 No. 2 No. 3 No. 4	105,100 16,100 79,000 24,400	—	—	105,100	3700
C. Journal	No. 1 No. 2 No. 3 No. 4	74,800 54,800 71,000 56,200	(1)	127,000 11,000 13,000 17,000	147,500	5200
D. Crank-pin No. 2	No. 1 No. 2 No. 3 No. 4	25,200 74,800 57,400 30,300	(1)	127,000 11,000 13,000 17,000	137,000	4820
E. Journal	No. 1 No. 2 No. 3 No. 4	62,600 80,600 80,600 62,600	(2)	112,000 115,000 31,000 31,000	140,000	4930

Position.	Which crank 30° past firing dead centre.	Bending moment in. lb. (see previous tables).	Number of twisting moment curve.	Twisting moment in. lb. from curves.	Equivalent twisting moment in. lb.	Maximum shear stress lb. per sq. in.
F. Crank-pin No. 3	No. 1	30,300		112,000		
	No. 2	57,400	(2)	115,000	128,500	4530
	No. 3	74,800		31,000		
	No. 4	25,200		31,000		
G. Journal	No. 1	56,200		100,000		
	No. 2	71,000	(3)	93,000	117,000	4120
	No. 3	54,800		93,000		
	No. 4	74,800		45,000		
H. Crank-pin No. 4	No. 1	24,400		100,000		
	No. 2	79,000	(3)	93,000	122,000	4300
	No. 3	16,100		93,000		
	No. 4	105,100		45,000		
I. Journal	No. 1			82,000		2890
	No. 2	—	(4)	82,000	—	
	No. 3			82,000		
	No. 4			82,000		

Conclusions.—Maximum Shear Stress, 5200 lb. sq. in.—Taking the fatigue stress in shear for mild steel, subject to combined bending and twisting at 15,000 lb. per sq. in., the factor of safety for a shaft newly lined up is about 3, and diminishes very considerably as the bearings become worn out of level.

The high values of the stresses at the centre of the shaft point to the advisability of making all couplings between sections of the crank-shaft of the full torsional strength of the shaft, i.e. the aggregate shearing area of the coupling bolts multiplied by the radius of their pitch circle should be equal to the twisting modulus of the shaft. Thickness of coupling flanges $^1/_4$ diameter of the shaft.

CHAPTER VI

FLY-WHEELS

The Functions of a Fly-wheel are :—

1. To keep the degree of uniformity within specified limits.

2. Where alternators running in parallel are in question to limit the angular advance or retardation of rotation to a specified fraction of a degree ahead of or behind an imaginary engine rotating with perfectly uniform angular speed.

3. To limit the momentary rise or fall in speed when full load is suddenly thrown off or on.

4. To facilitate starting under compressed air.

In addition to the above the fly-wheel usually serves as a barring or turning wheel and a valve setting disc ; also the inertia of the fly-wheel has great influence in determining the critical speed at which torsional oscillations of the crank-shaft are set up.

Fly-wheel Effect.—The fly-wheel effect of a rotating body is its polar moment of inertia (mass $\times$ radius of gyration squared) about its axis of rotation. For a fly-wheel or pulley it is found approximately by multiplying the weight of the rim in pounds by the square of the distance in inches from the axis to the centre of gravity of the section of the rim, the result being in in.^{2} lb. units. This underestimates the moment of inertia slightly, and a more accurate method will be described later. The fly-wheel effect of the running gear of one cylinder is found with sufficient accuracy for most purposes by adding the weight of the revolving parts (crank-pin plus unbalanced part of two crank webs plus 0·65 of the connecting rod) to half the weight of the reciprocating parts (0·35 of the connecting rod plus cross-head plus piston-rod plus piston, etc.), and multiplying the sum by the square of the crank radius.

For a screw propeller the radius of gyration may be taken as 0·35 of the extreme radius if details are not available.

Degree of Uniformity.—

$$\text{Degree of uniformity} = \frac{\text{Max. speed} - \text{Min. speed}}{\text{Mean speed.}}$$

Let d = Degree of uniformity.

w_1 = Max. angular speed in radians per second.

w_2 = Min. ,, ,, ,, ,, ,,

Then $d = 2\dfrac{(w_1 - w_2)}{(w_1 + w_2)}$ ——(1)

For a specified value of "d" the necessary fly-wheel effect is calculated by means of the resultant twisting moment curve of the engine. Let Fig. 49 represent the twisting moment curve and the line A C the mean twisting moment. Let A B C be the loop of largest area (with multi-cylinder engines there are in general as many positive and negative loops in a complete cycle as there are cylinders, and the area of each positive loop is the same as that of each negative loop). If the loop A B C is above the line A C, then the speed of the engine is a minimum at A and a maximum at C, and the increase of rotational energy of the fly-wheel, etc., between A and C is equal to the work represented by the area of the loop A B C.

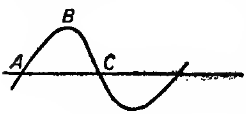


FIG. 49.

Let A = Area of loop A B C in sq. in. on the diagram.

E = Work represented by A B C in in. lb.

a = Scale to which turning moments are plotted in in. lb. to the inch.

b = Scale to which crank-shaft degrees are plotted in degrees to the inch.

Then $E = \dfrac{A \times a \times b}{57 \cdot 3}$, 57·3 being the number of degrees in a radian.

Let WK^2 = Fly-wheel effect (moment of inertia) in in.^{2} lb.

Then kinetic energy of wheel $= \dfrac{WK^2 . w^2}{2\,g}$ ($g = 386$ in. /sec.^{2}).

Change of kinetic energy from A to C

$$= \frac{WK^2}{2\,g}(w_1^2 - w_2^2) = \frac{WK^2}{2\,g}(w_1 - w_2)\,(w_1 + w_2) = \frac{WK^2 . d}{4\,g}(w_1 + w_2)^2$$

$$= WK^2 . d . w^2(\text{MEAN}) \div g$$

if the difference between w_1 and w_2 is small.

But the change of kinetic energy is equal to E.

$$\therefore \quad E=\frac{WK^2}{386}.d.w^2$$

and $d=\frac{E\times 386}{WK^2.w^2}$ or $WK^2=\frac{E\times 386}{w^2.d}$ ——(2)

Example : Single cylinder engine 10″ bore×15″ stroke. Revs. 300. Turning moment diagram as in Fig. 48, full line. E=151,000 in. lb. Radius of gyration of wheel 30″. Required to find the weight of the wheel to give a degree of uniformity of 1/80.

$$w=\frac{300\times 2\pi}{60}=31\cdot 4 \text{ radians per sec.}$$

$$W.K^2=\frac{E\times 386}{w^2\cdot d}=\frac{151{,}000\times 80\times 386}{31\cdot 4^2}=4{,}730{,}000$$

Fly-wheel effect of running gear $\left(267+\frac{236}{2}\right)\times 7\cdot 5^2=21{,}600$ in.^{2} lb.

WK^2 for fly-wheel $=4{,}730{,}000-21{,}600=4{,}708{,}400$ in.^{2} lb.

but K=30″

$$\therefore \quad W=\frac{4{,}708{,}400}{30^2}=5230 \text{ lb.}=2\cdot 34 \text{ tons.}$$

Twisting Moment Diagrams for two and four stroke engines having from one to eight cylinders are shewn in Figs. 50 to 61. These have been drawn for an engine 10″ bore by 15″ stroke. As the twisting moments of two engines of different sizes are proportional to the bore^2×stroke, these curves may be used for engines of any size by multiplying the moments by the bore^2 (in inches^2)×the stroke (in inches) and dividing by 1500. The excess energy represented by the largest loop in each diagram is given in the schedule below for each case.

	Four Stroke Engines.		Two Stroke Engines.	
Number of Cylinders.	E in in. lb. for 10″×15″ Cylinder.	E in in. lb. for 1″×1″ Cylinder.	E in in. lb. for 10″×15″ Cylinder.	E in in. lb. for 1″×1″ Cylinder.
1	151,000	101·0	125,500	83·7
2	127,500	84·8	58,500	39·0
3	87,300	58·2	48,700	32·4
4	38,500	25·7	39,000	26·0
6	39,100	26·1	11,150	7·4
8	31,700	21·1	2,200	1·5

Substituting those values of E for a cylinder 1 in. × 1 in. in equation (2) the following formula is obtained :—

$$WK^2 = \frac{C.B^2.S}{d\left(\frac{n}{100}\right)^2}$$

Where B = Bore of cylinder in inches.
S = Stroke in inches.
n = Revolutions per minute.

Values of C are given in the following schedule :—

No. of Cylinders.	C for 4 Stroke Engine.	C for 2 Stroke Engine.
1	355	243
2	298	137
3	204	114
4	90	91
6	91	26
8	74	5

Values for "d" used in Diesel Engine Practice.—For certain purposes, as for instance spinning mills, a fine degree of uniformity is desirable, and d = about $\frac{1}{100}$.

For direct coupled continuous current dynamos d = $\frac{1}{80}$ is sufficiently fine to prevent flickering of lights, and may be used unless considerations of momentary governing demand a heavier wheel than the use of this figure would give rise to.

For marine engines and land drives, where regularity of turning is not of importance, "d" may be about $\frac{1}{10}$.

The above values for the degree of uniformity must be used with caution, as in a large number of cases (particularly four stroke engines of six cylinders and upwards and two stroke engines of three cylinders and upwards) the considerations discussed in the next article outweigh those of regularity in turning.

Momentary Governing.—Under the head of governing it is usually specified that the rise in speed when the load is thrown off suddenly or the fall in speed when the load is suddenly thrown on shall not exceed a certain percentage (usually between 5 and 12) of the mean speed. Actually the governor

has relatively small control over this rise or fall of speed, as at the instant when the load is thrown off sufficient fuel has already been deposited in the pulverisers to carry the engine against full load for a period which may be anything up to

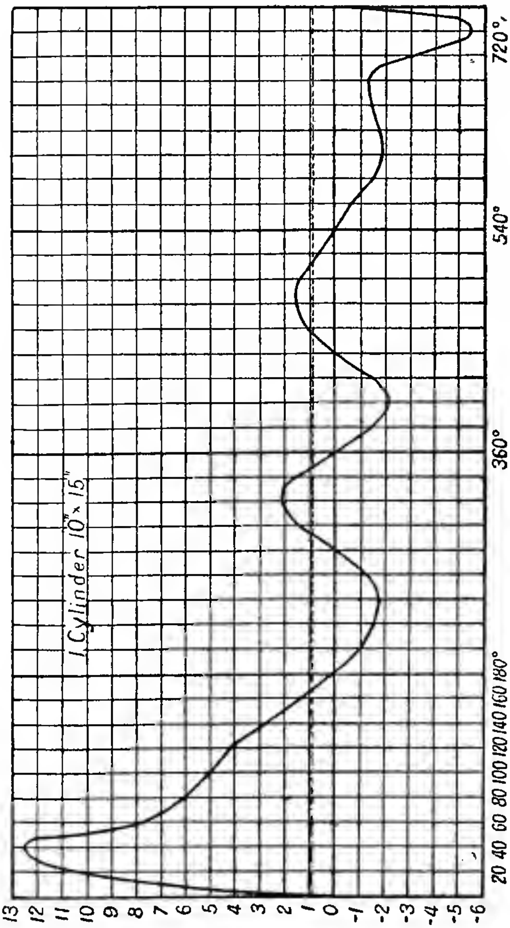


Fig. 50.

FOUR STROKE ENGINE

two revolutions in the case of a four stroke engine. The brake energy developed during this period is entirely devoted to accelerating the fly-wheel and other rotating masses. Owing to the fact that the governor does not act immediately the load is thrown off, the wheel should be capable of absorbing the

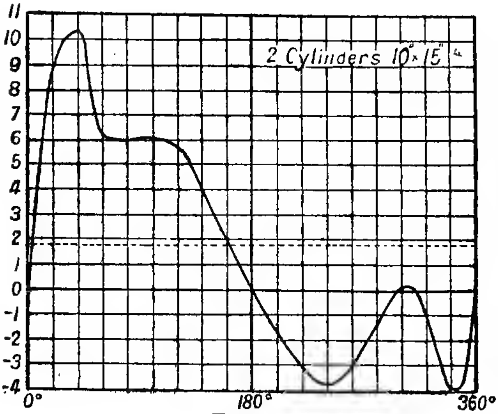


FIG. 51.

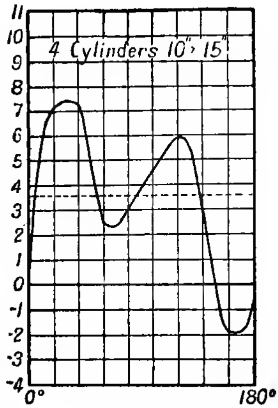


FIG. 52.

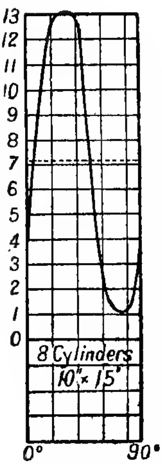


FIG. 53.

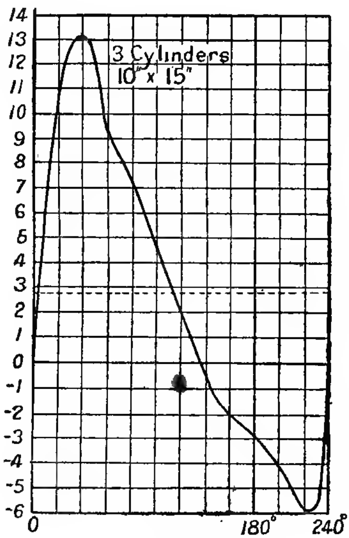


FIG. 54.

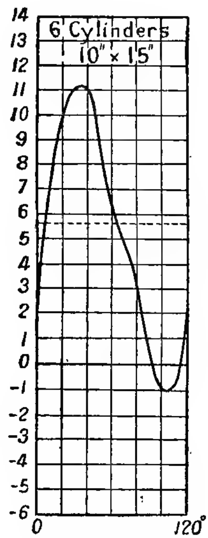


FIG. 55.

FOUR STROKE ENGINES.

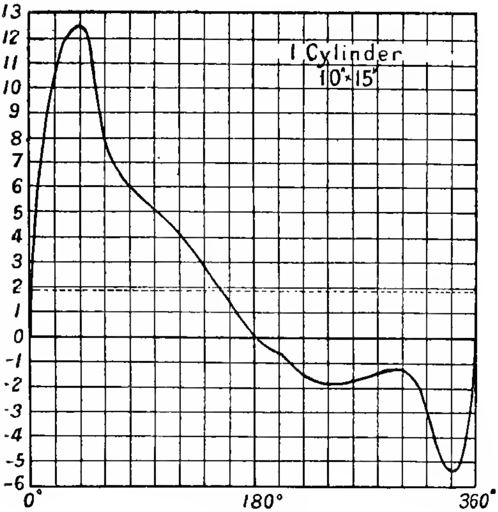


Fig. 56.

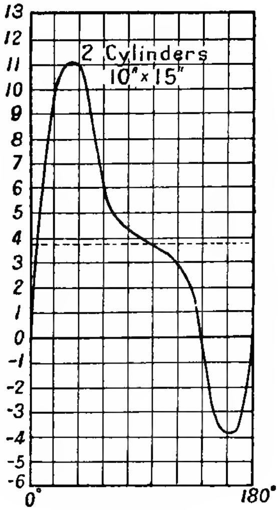


Fig. 57.

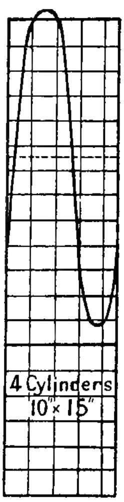


Fig. 58.

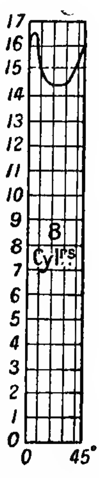


Fig. 59.

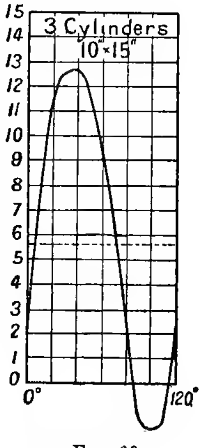


Fig. 60.

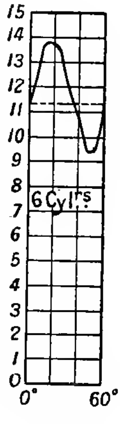


Fig. 61.

TWO STROKE ENGINES.

whole power of the engine for about three revolutions in the case of a four stroke engine, about 1·5 of a revolution in the case of a two stroke engine.

Example :	B.H.P. of engine (four stroke) .	180
	Revolutions per minute . .	375
	Momentary rise in speed when full load is suddenly thrown off	12%
	Radius of gyration of wheel .	18 in.

It is required to find the weight of the fly-wheel, neglecting the fly-wheel effect of the running gear.

Work done per revolution at full load $\frac{180 \times 33{,}000 \times 12}{375}$ in. lb.

Energy corresponding to three revolutions

$$=\frac{180 \times 33{,}000 \times 36}{375}=570{,}000 \text{ in. lb.}$$

Angular speed at full load $\frac{2\pi \times 375}{60}=39{\cdot}3$ radians per second.

Momentary angular speed when load is suddenly thrown off

$$39{\cdot}3 \times 1{\cdot}12=44{\cdot}0 \text{ radians per second.}$$

If W = weight of wheel, then :—

$$\frac{W \times 18^2}{2 \times 386}(44^2-39{\cdot}3^2)=570{,}000 \text{ in.}^2 \text{ lb.}$$

and W = 3470 lb.

Alternators in Parallel.—When two or more alternator sets are being run in parallel it is a necessary condition for working that they keep almost exactly in phase. Due to inequalities of twisting moment, slight differences of phase inevitably occur, and these give rise to synchronising currents between the various machines, the tendency of these currents being to accelerate the lagging machines and retard the leading ones. This effect keeps the whole system in a state of stability, but cannot be relied on to correct any large fluctuations, and on this account it is usual to specify that the maximum deviation from uniform rotation shall not exceed three electrical degrees on either side of the mean. If the alternator under consideration has a field of two poles only, then the electrical degrees correspond to crank-shaft degrees. In general, if the number of pole pairs is " p," then one crank-shaft degree corresponds

to "p" electrical degrees. So far as the engine designer is concerned, then, the problem consists in ascertaining the fly-wheel effect required to keep the cyclic fluctuations on the engine fly-wheel within a certain number of degrees, or more commonly within a certain fraction of a degree of revolution, on either side of the mean.

The method of calculation may be described briefly thus :—

(1) Assume any convenient figure for the fly-wheel effect, e.g. 100,000 in.^{2} lb.

(2) Plot twisting moment curve for complete period, taking the zero of ordinates at the mean twisting moment.

(3) Reduce crank angles to time in seconds, assuming uniform rotation.

(4) Reduce twisting moments to angular acceleration in degrees per second^2 by dividing by the assumed fly-wheel effect and by the acceleration due to gravity (386 in. per sec.^{2}) and multiplying by the number of degrees in a radian (57·3).

(5) Plot angular acceleration to time or crank angle base.

(6) Integrate by planimeter, or otherwise, obtaining angular speed curve.

(7) Integrate again, obtaining angular displacement curve.

(8) Measure maximum deviation from the mean position in degrees.

(9) Increase or decrease the assumed fly-wheel effect in proportion as the angular deviation so found is more or less than the deviation specified. This gives the fly-wheel effect required.

Example : Three cylinder, four stroke engine :—

Bore	20 in.
Stroke	32 ,,
Revolutions per minute . .	150
Number of pole pairs . . .	20
Angular deviation.	3 electrical deg.

Twisting moment curve as in Fig. 62. The mean twisting moment being taken as the basis.

The whole calculation is contained in the table below, in conjunction with Figs. 62, 63 and 64.

Since the engine makes 150 revolutions per minute, therefore

$$20^\circ = \frac{20 \times 60}{150 \times 360} = 0{\cdot}0222 \text{ sec.}$$

Assume fly-wheel effect of 10^6 in.^{2} lb. for purposes of calculation. Then :—

$$\text{Acceleration in radians per sec.}^2 = \frac{\text{Twisting moment} \times 386}{10^6}$$

$$\text{in degrees per sec.}^2 = \frac{\text{T.M.} \times 57{\cdot}3 \times 386}{10^6} = \frac{\text{T.M.}}{45{\cdot}3}$$

Referring to the table below :—

Values given in column 2 are scaled off Fig. 62.

„ „ 3 are obtained by dividing those in column 2 by 45·3.

„ „ 4 are obtained by multiplying the values in column 3 by 0·0222 sec.

Column 5 is obtained by successive addition of speed increments.

Column 6 contains corrections necessitated by the fact that the resultant of column 5 is not zero, owing to errors.

Column 7 gives corrected speeds which are plotted in Fig. 63.

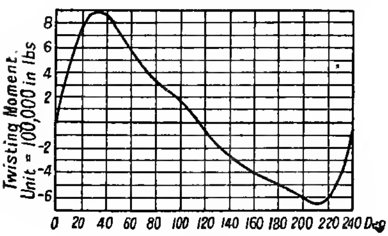


Fig. 62.

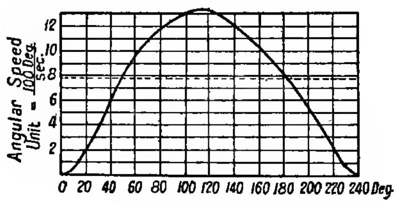


Fig. 63.

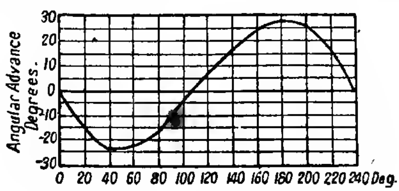


Fig. 64.

Columns 8, 9 and 10 are obtained similarly to columns 2, 4 and 5.

Total swing in phase (see Fig. 64) $24+28=52°$ or $26°$ each side of the mean.

Degrees.	Average Twisting moment over 20° Inlb. ÷ 1000.	Average Acceleration over 20°. Deg./sec.²	Increase or Decrease in Speed. Deg./sec.	Summation of Speed Increments. Deg./sec.	Correction. Deg./sec.	Corrected Speed. Deg./sec.	Average Speed above or below Mean Speed. Deg./sec.	Increase or Decrease of Angle. Deg.	Summation of Angular Advances or Retardations. Deg.
0				0	0	0			0
	440	9710	216				—700	—15·5	
20				216	9	207			—15·5
	870	19,200	426				—340	—7·5	
40				642	18	624			—23·0
	730	16,200	360				40	0·9	
60				1002	27	975			—22·1
	440	9710	216				300	6·7	
80				1218	35	1183			—15·4
	270	5960	132				480	10·7	
100				1350	44	1306			—4·7
	30	662	15				530	11·8	
120				1365	53	1310			7·1
	—180	—3980	—88				470	10·4	
140				1277	62	1215			17·5
	—340	—7500	—167				330	7·3	
160				1110	71	1039			24·8
	—430	—9480	—211				150	3·3	
180				899	80	819			28·1
	—540	—11,900	—264				—90	—2·0	
200				635	88	547			26·1
	—640	—14,100	—313				—390	—8·7	
220				322	97	225			17·4
	—440	—9710	—216				—700	—15·5	
240									1·9

Allowable swing, 3 electrical deg. $=\frac{3}{20}=$ crank-shaft deg.

Fly-wheel effect assumed for calculation, 1,000,000 in.^{2} lb.

Therefore fly-wheel effect required

$$=\frac{1{,}000{,}000\times 20\times 26}{3}=173\times 10^6 \text{ in.}^2 \text{ lb.}$$

If radius of gyration of wheel is 65 in.

$$\text{Weight of wheel}=\frac{173\times 10^6}{65^2\times 2240}=18{\cdot}3 \text{ tons.}$$

Allowance for the fly-wheel effect of the alternator rotor would reduce this figure a little.

Torsional Oscillations and Critical Speeds.—In the great majority of Diesel Engines the critical speed at which torsional oscillations of the crank-shaft would occur lies far above the practical range of the engine. Serious cases do arise occasionally, however, and will arise more frequently when high speed two stroke engines of six cylinders and upwards become more common. The trouble when it arises is generally disposed of very simply by altering the fly-wheel effect. Supposing for the moment that a marine engine is under consideration. Then the fly-wheel, the crank masses, the propeller and the shafting, etc., constitute an elastic system having a natural frequency of torsional oscillations which depends on the amounts and positions of the fly-wheel effects of its component parts and on the stiffness of the shafting. If this natural frequency happens to be the same as the frequency of the torsional impulses due to the working strokes of the engine, then the oscillations tend to become accumulative, vibrations are felt in the shafting, and the crank-shaft may hammer in its bearings. Then the engine is running at a critical speed. In general a two stroke engine gives as many torsional impulses per revolution as there are cylinders and a four stroke engine half this number, so that a two stroke engine attains its critical speed at one-half the number of revolutions required by a four stroke engine. For example : Suppose the crank-shaft, etc., of a six cylinder four stroke engine has a natural frequency of 2400 complete oscillations per minute, then the critical speed will be 800 revolutions per minute. A similar engine working on the two stroke cycle would have a critical speed of 400 revolutions per minute. It will be obvious that all the revolving masses in connection with the crank-shaft (apart from trifling items) must be taken into

consideration, so that the critical speed of a marine engine coupled to a dynamo for testing purposes will be different to that obtained when the engine is installed in the ship.

Natural Frequency of Torsional Oscillation.—Consider the simple system shewn in Fig. 65, consisting of a shaft fixed at one end and carrying a fly-wheel at the other. If the fly-wheel be turned through an angle against the torsional resistance of the shaft and then released suddenly the system will oscillate until the energy has been dissipated in friction, the angle through which any section of the shaft oscillates being proportional to the distance from the fixed end. The latter is called the Node.

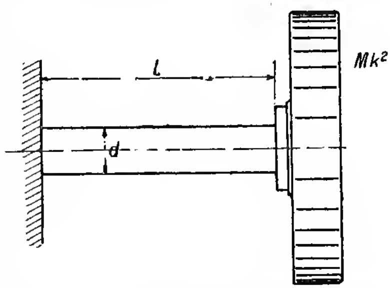


FIG. 65.

Let f = Frequency in complete oscillations per second.
F = Frequency ,, ,, ,, minute.
d = Diameter of the shaft in inches.
I = Polar moment of inertia of shaft section in inches^4.
l = Length of shaft in inches.
W = Weight of the wheel in lb.
K = Radius of gyration of the wheel in inches.
g = Acceleration due to gravity in inches per second^2 (386)
G = Modulus of rigidity of the shaft material (about 12,000,000 lb. per sq. in.).

Then :—

$$f=\frac{1}{2\pi}\sqrt{\frac{I.G.g}{l.W.K^2}} \text{ and } F=9{\cdot}55\sqrt{\frac{I.G.g}{l.W.K^2}} \text{——(1)}$$

If the shaft is not of the same diameter throughout its length, but consists of sections of length l_1, l_2, etc., of diameter d_1, d_2, etc., then the equivalent length l_0 of shift of standard diameter d_0 is given by

$$l_0=l_1\left(\frac{d_0}{d_1}\right)^4+l_2\left(\frac{d_0}{d_2}\right)^4+\ldots \text{ etc.}$$

For example : 1 ft. of 6 in. shafting is equivalent to 16 ft. of 12 in. shafting, 16 ft. being $\left(\frac{12}{6}\right)^4$. For this reason, the lengths occupied by coupling flanges, etc., are negligible.

If there are a number of fly-wheels or other rotating masses at different distances from the node, then the frequency is given very closely by the following :—

$$F=9\cdot55\sqrt{\frac{I.G.g}{l_1W_1K_1^2+l_2W_2K_2^2+\ldots\text{etc.}}}\quad\text{——(2)}$$

In the case of an engine and shafting no point on the latter is fixed, and the position of the node is to be inferred from considerations of dynamic equilibrium. It will be obvious that a uniform shaft with an equal wheel at each end will have its node at the centre, the oscillations of the two wheels being equal in magnitude and opposite in sense at every instant. It is almost equally apparent that in a similar case, with unequal wheels the position of the node will divide the shaft into two lengths in inverse ratio to the fly-wheel effects at their ends.

In general, the position of the node is determined with sufficient accuracy to enable the critical speed to be predicted within a few revolutions per minute by treating the fly-wheel effects (moments of inertia) as though they were weights and locating the node at their centre of gravity.

A crank-shaft may be treated as a uniform shaft of the diameter of the journals.

Example : Six cylinder, two stroke marine engine.

Bore.	24 in.
Stroke	35 in.
Revolutions per minute	120
Diameter of shaft	15 in.
Lengths of crank-shaft	as in Fig. 66
Weight of fly-wheel	12,000 lb.
Radius of gyration of fly-wheel	40 in.
Weight of revolving parts for one crank .	4,500 lb.
,, reciprocating parts	51,000 lb.
,, propeller	10,000 lb.
Radius of gyration of propeller	20 in.
Shafting	as in Fig. 66

Reduction of Shafting to Standard Diameter of 15 Inches.—

Length of 14″ shafting . . . . . . 620 in.

Equivalent length of 15″ shafting, $620\left(\frac{15}{14}\right)^4=815$ in.

Equivalent length of 15″ shafting between the propeller and fly-wheel $=150+815+60=1025$ in.

Moment of inertia (polar) of 15″ shaft $=\frac{\pi}{32}\times 15^4=49{,}600$ in.^{4}

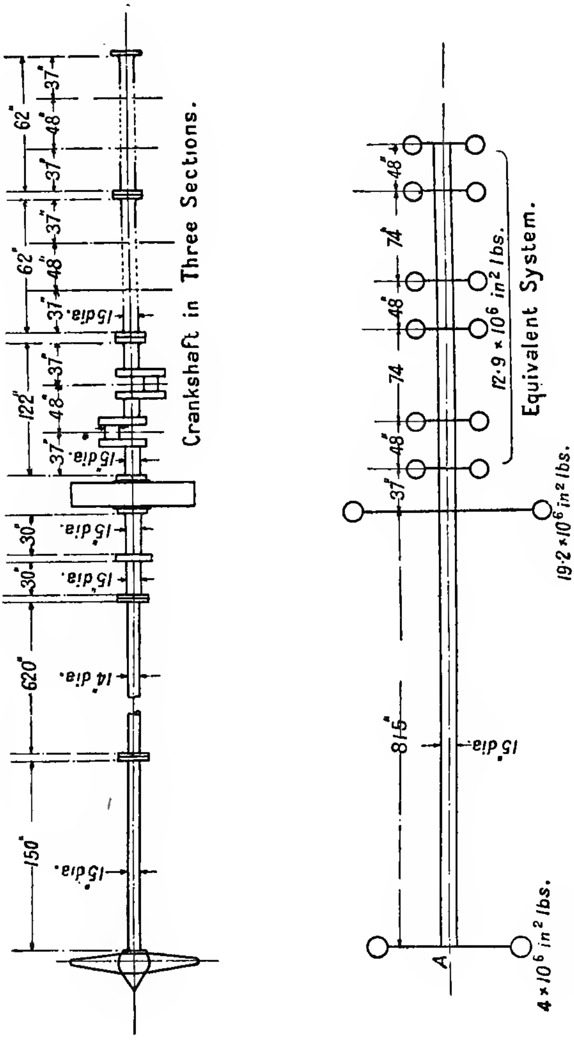


FIG. 66.

Fly-wheel Effects.—

$W.K^2$ for propeller $=10,000 \times 20^2 = 4,000,000$ in. lb.

$W.K^2$ for crank masses $=(4500+2500)\ 6 \times 17{\cdot}5^2$
$= 12,900,000$ in.^{2} lb.

$W.K^2$ for fly-wheel $=12,000 \times 40^2 = 19,200,000$ in.^{2} lb.

Position of Node.—Take moments of fly-wheel effects about A :—

$$\begin{array}{rr} 19{\cdot}20 \times 1025 = & 19,700 \\ 12{\cdot}9\ (1025+37+48+74+24) = & 15,600 \\ \hline \text{Total} & \underline{\underline{35,300}} \end{array}$$

The distance of the node from $A = \dfrac{35,300}{36{\cdot}1} = 978$ in.

Dealing with the part of the system to the left of the node and applying equation (1) :—

$$F = 9{\cdot}55 \sqrt{\frac{49,600 \times 12 \times 10^6 \times 386}{978 \times 4 \times 10^6}} = 2320 \text{ oscillations per minute.}$$

And the critical speed $\dfrac{2320}{6} = 386$ R.P.M.

which is far above the working range of the engine.

Critical speeds of the second and third order, and so on, are possible at 193, 128, 96 R.P.M., etc. ; but these are not, as a rule, important.

It is sometimes useful to repeat the calculation for different weights of fly-wheel and plot a curve connecting moment of inertia of fly-wheel and critical speed, so that the possible variation of critical speed obtainable by altering the fly-wheel can be seen at a glance.

Torsional oscillations about two or more nodes are also possible, but as these involve higher speeds, and are therefore more subject to damping, it is doubtful if they are of much practical importance.

To find the Moment of Inertia of a Fly-wheel.—In the first instance, suppose the wheel in question is a disc wheel, i.e. a solid of revolution. Referring to Fig. 67, the thick, full line represents the section of the wheel. Z Z is the axis and S S is a line through the extreme radius of the wheel parallel to the axis at a distance R from the latter. Rule any line A B parallel to the axis, cutting the outline of the section in A and B. Project A and B on to S S at C and D. Join C and D to any

convenient point O on the axis, cutting A B in A_1 and B_1. Proceed similarly with different positions of the line A B and join up the various positions of A_1 and B_1, thus obtaining a new figure—the First Derived Figure. Treat this figure as though it were the original figure, and obtain the Second Derived Figure. Similarly with this figure obtaining the Third Derived Figure.

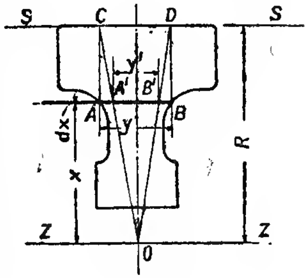


FIG. 67.

Let A = Area of original section in sq. in.
A_1 = Area of First Derived Figure in sq. in.
A_2 = Area of Second Derived Figure in sq. in.
A_3 = Area of Third Derived Figure in sq. in.
w = Weight in lb. of one cubic inch of the material.

$$\text{Then } A=\int_0^R y.dx, \quad A_1=\frac{1}{R}\int_0^R x.y.dx, \quad A_2=\frac{1}{R^2}\int_0^R x^2.y.dx,$$

$$A_3=\frac{1}{R_3}\int_0^R x^3.y.dx.$$

$$\text{Weight of wheel}=2\pi.w\int_0^R x.y.dx=2\pi.w.R.A_1$$

$$\text{Moment of inertia of wheel}=2\pi w\int_0^R x^3y.dx=2\pi w\,R^3A_3$$

$$\text{Radius of gyration}^2=R^2.\frac{A_3}{A_1}$$

The above hold good for any position of the line S S, which may therefore be taken where most convenient. In cases where the section tapers towards the extreme radius (a screw propeller, for instance) the line S S is best located at a distance of about one-half or one-third of the extreme radius from the axis.

In the case of a screw propeller or a fly-wheel with arms, the rotating body must first be reduced to an equivalent disc wheel. This is readily done as follows: Describe a radius R which cuts through the arms or blades, as the case may be. Divide the total area of section at this radius by $2\pi.R$ and the result is the thickness of the equivalent disc at this radius. Repeat for a number of different radii covering the whole range.

Types of Fly-wheels.—Fig. 68 shews a disc wheel cast in one

piece and provided with a number of drilled holes in the rim for turning the engine by means of a bar. Degree marks are cut on the edge of the rim to facilitate valve setting. Fig. 69 shews a disc wheel with a separate centre, an arrangement which

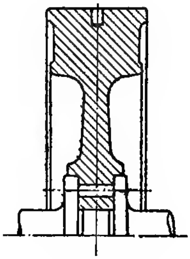


Fig. 68.

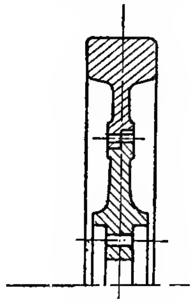


Fig. 69.

makes it easier to obtain a sound casting. Large wheels are usually cast in two pieces, and Fig. 70 shews a design which is suitable for weights up to at least 20 tons. It should be noted that no keys are provided for securing the wheel to the shaft. If the boss of the wheel is bored one-thousandth per inch of

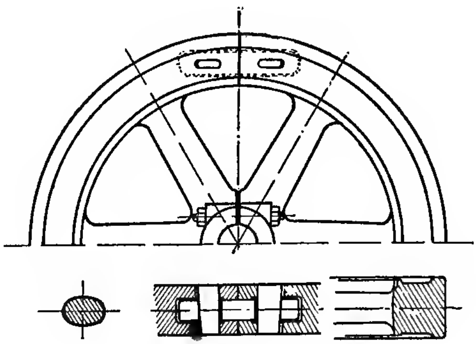


Fig. 70.

diameter less than the shaft and the bolts are drawn up at about the temperature of boiling water the frictional grip is quite sufficient for the largest wheels and the danger of splitting the boss of the wheel involved in the use of keys is avoided. The same applies to pulleys for belt or rope drives. A rather more elaborate wheel, in which greater precautions have been

taken, is shewn in Fig. 71. In this case it is advisable to make the bore of the wheel the same as the shaft diameter and to give a shrinking allowance of about one-thousandth per inch of diameter to the bore of the shrunk ring. For large stationary engines some form of barring gear is necessary, and where electric power is available a motor-driven gear is a great convenience. For marine engines a worm, or other self-locking gear, is essential, and where the auxiliaries are electrically driven an electric turning gear should be fitted, as the use of the latter greatly expedites adjustments to the valve gear.

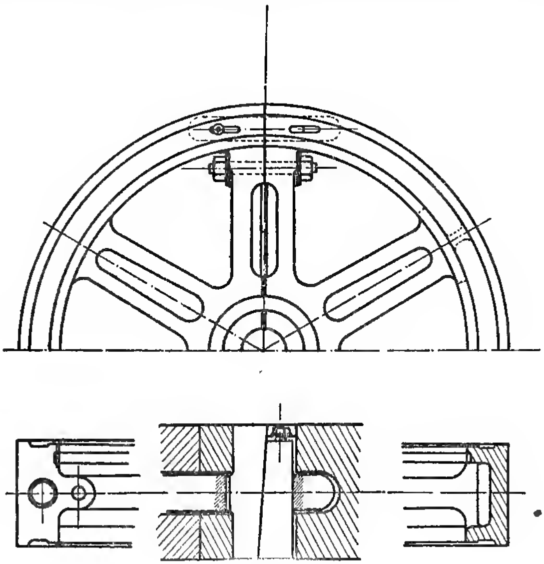


FIG. 71.

Strength of Fly-wheels.—Continental practice favours a peripheral speed of about 100 feet per second for cast iron fly-wheels, and this corresponds to a stress of about 1000 lb. per sq. in. An investigation by Mr. P. H. Smith into the case of a split fly-wheel which burst at Maidenhead in 1912 shewed that the engine (the fly-wheel of which had a normal working peripheral speed of about 100 feet per second) was running about double its normal speed, and as the stress varies as the square of the speed, it follows that the factor of safety under normal conditions was about 4. Destruction tests of wheels and models of wheels shew that the bursting speed is about 200 feet

per second for split wheels and 400 feet per second for solid wheels. The discrepancy seems very large and difficult to account for. Average British practice is in favour of a slightly lower peripheral speed (about 90 feet per second). With marine engines considerations of space generally necessitate a still lower figure. The strength calculations for a fly-wheel will be illustrated by an example.

Example: Required to find the approximate dimensions of a fly-wheel suitable for 180 revolutions per minute given that

$$W.K^2 = 40{,}000{,}000 \text{ in.}^2 \text{ lb.}$$

Peripheral speed, 100 ft. per sec.

Maximum twisting moment due to engine, 450,000 in. lb.

Outside radius of wheel $= \dfrac{100 \times 12 \times 60}{2\pi \times 180} = 63{\cdot}7$ in, say 64 in.

Take inside radius of rim $=52$ in.

Then radius of C.G. of rim section $=58$ in.

Let $B =$ width of rim. Then :—

$$\text{Weight of rim} = 12 \times B \times 2\pi \times 58 \times 0{\cdot}26$$

And approximate moment of inertia

$$= 12 \times B \times 1{\cdot}64 \times 58^3 = 40{,}000{,}000 \text{ in.}^2/\text{lb.}$$

$$\text{From which } B = 10{\cdot}5 \text{ in.}$$

Since the stress due to a peripheral speed of 100 ft. per sec. is about 1000 lbs./in.^{2}, the total tension at each joint of the rim is equal to $12 \times 10{\cdot}5 \times 1000 = 126{,}000$ lb., for which pull the dowel and cotter section must be designed.

Allowable stress in dowel, say 6000 lb. per sq. in.

Effective area of dowel section $\dfrac{126{,}000}{6000} = 21$ sq. in.

Since about one-third of the section of the dowel is cut away by the cotter hole (see Fig. 72), the gross sectional area of the dowel must be $\dfrac{21 \times 3}{2} = 31{\cdot}5$ sq. in., say 4 in. $\times$ 8 in.

Thickness of cotter $= \dfrac{8}{3} =$ about $2\frac{3}{4}$ in.

Bearing pressure of cotter on dowel

$$\frac{126{,}000}{2{\cdot}75 \times 4} = 11{,}450 \text{ lb. per sq. in.}$$

which is allowable.

If the hole for the dowel is made $\frac{1}{2}$ in. wider than the dowel itself, the bearing length for the cotter on the rim will be $10{\cdot}5 - 4{\cdot}5 = 6$ in., and bearing pressure of cotter on rim

$$\frac{126{,}000}{2 \cdot 75 \times 6} = 7650 \text{ lb. per sq. in., also allowable.}$$

Allowable shear stress for cotter (which is in double shear), say 5000 lb. per sq. in.

Depth of cotter $\dfrac{126{,}000}{5000 \times 2 \cdot 75} = 9 \cdot 2$ in., say $9\frac{1}{2}$ in. over the rounded ends.

The distance " l " between the inside edge of the cotter hole and the rim joint must be sufficient to obviate risk of the

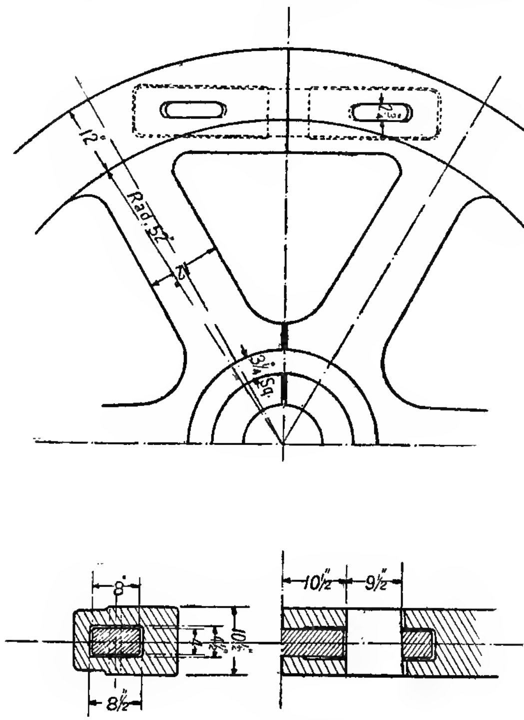


FIG. 72.

intervening metal being torn out in double shear. (This point is sometimes overlooked in otherwise well-proportioned rim joints.)

Allowing a shear stress of 1000 lb. per sq. in. :—

$$l = \frac{126{,}000}{1000 \times 2 \times 6} = 10 \cdot 5 \text{ in.}$$

Owing to the difficulty of analysing the straining actions on the arms it is well to give the latter ample proportions.

An approximate method of calculation is given below.

Assume that the maximum twisting moment due to the engine (450,000 in. lb.) is transmitted to the rim by means of a constant shear force across the arms, and that the bending moment is a maximum at each end of an arm and zero at the centre.

The length of each arm from boss to rims is about 40 in., and the distance of its centre from the centre of the wheel about 32 in.

Then shear force in each arm $= \frac{450,000}{6 \times 32} = 2340$ lb. (assuming six arms).

And maximum bending moment at end of each arm $= 2340 \times 20 = 46,800$ in. lb.

Taking a low stress of 500 per sq. in., to allow for direct tension in the arms, bending modulus of arm section

$$= \frac{46,800}{500} = 93 \text{ in.}^3$$

This is satisfied by a rectangle section 6 in. $\times$ 10 in., which could be replaced by an oval section about 7 in. $\times$ 12 in. to reduce wind resistance. The bolts at the hub of the wheel are sometimes made as strong as the rim joint, in which case the core area of two bolts will be the same as the net effective area of one dowel, viz., 21 sq. in.

This gives a bolt of about 4 in. diameter. If shrunk rings are employed these will have a square section about $10 \cdot 5 = (3\frac{1}{4} \text{ in.})^2$.

The above calculations must be regarded as preliminary only, and give the draughtsman a basis on which to start designing. The next step will be to check over the weight and radius of gyration of the complete wheel in the manner already describe[d]. The dimensions and stress calculations will then be amende accordingly.

Literature.—For information on the strength of fly-wheel , see :—Unwin, W. C., and Mellanby, A. L., "The Elements of Machine Design," Part II.

For information on critical speeds, see :—Morley, A., "Critical Speeds for Torsional and Longitudinal Vibrations," *Engineering*, December 9th, 1910.

CHAPTER VII

FRAMEWORK

A LARGE number of different types of framework have been employed in Diesel Engine construction, and a complete classification will not be attempted here. The outstanding types in successful practice may, however, be broadly divided into a few well-defined classes, as under :—

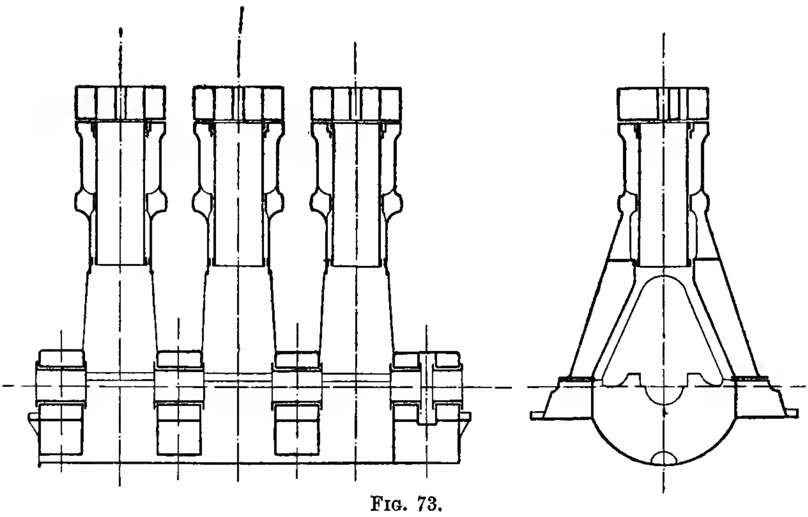


FIG. 73.

"A" Frame Type.—This is the earliest type of Diesel Engine construction, and on account of its merits is still very extensively used. Referring to the diagrammatic drawing Fig. 73, it will be seen that a stiff bedplate of box section is provided, and that each cylinder stands on its own legs without support from its neighbours. The legs of the column are cast integrally with the cylinder jacket, into which a liner is fitted. The breech end of the cylinder is closed by means of a deep cylinder cover of box section.

The main tensile load due to the cylinder pressure is transmitted from the cover through the jacket and legs to the bedplate. The reaction corresponding to this load occurs, of course, at the main bearings, and consequently that part of the bedplate between the column feet and the main bearing housings must be designed to deal with the bending moment occasioned by the fact that the tensile load in the columns and the reaction at the bearings are not in the same plane. Casting the cylinder jacket and column in one piece reduces fitting and machining operations to a minimum and the independence of the individual cylinders would appear to have no disadvantages so far as land engines are concerned.

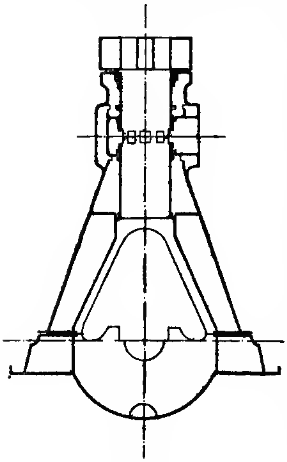


FIG. 74.

Fig. 74 shews the same type of construction applied to a two stroke land engine and Fig. 77 to a four stroke marine cylinder.

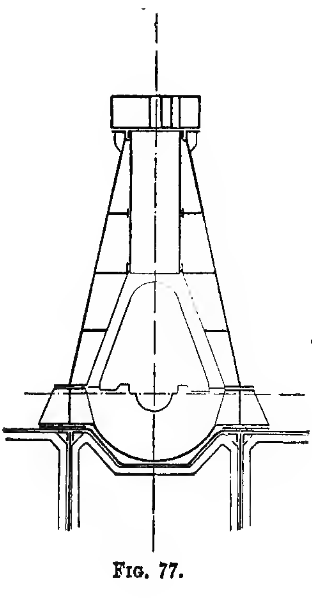


FIG. 77.

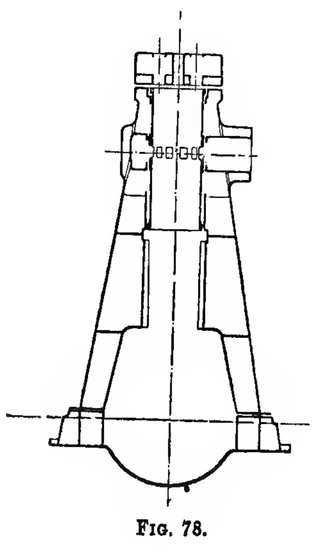


FIG. 78.

Slightly lengthening the column legs enables crosshead and guides to be fitted (see Fig. 78, which represents a large two cycle land engine). Occasionally one of the column legs takes the form of a steel tie rod, with a view to giving greater accessibility to the running gear and to enable the crank-shaft to be replaced, if necessary, without dismantling the whole engine. Unfortunately this arrangement nullifies many advantages of the "A" frame construction, as special splash guards must now be fitted to retain the lubricating oil, which office they do not always perform very efficiently, and also additional

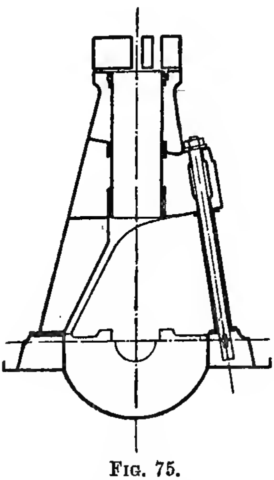


Fig. 75.

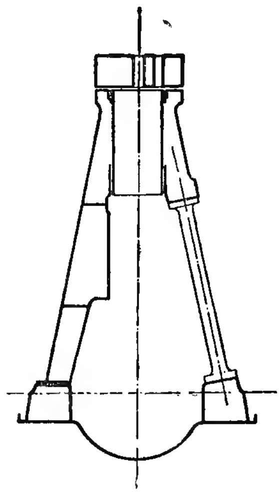


Fig. 76.

machining and fitting operations are introduced which add to the cost of production without increasing the efficiency of working. Figs. 75 and 76 shew this construction applied to trunk and crosshead engines respectively.

Crank-case Type.—The crank-case type of Diesel Engine was introduced when a desire was felt for higher speeds, necessitating forced lubrication. The crank-case bears external resemblance to that of a high speed steam engine (see Fig. 79). On the other hand, the high pressures dealt with in the cylinder of a Diesel Engine necessitate the crank-case being strengthened internally to an extent which is not found necessary in steam practice. Sometimes the box or girder construction of the crank-case is relied upon to transmit the tensile stresses from

the cover to the bedplate; more frequently, however, steel staybolts are provided for this purpose (see Fig. 80). The latter procedure, however, does not justify flimsy construction

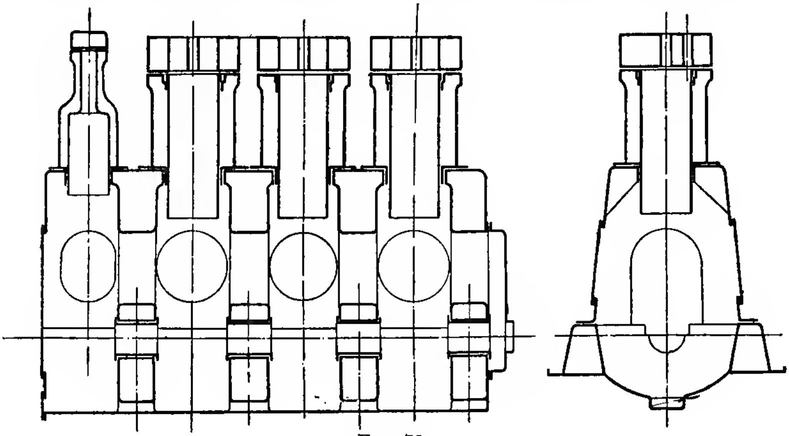


FIG. 79.

or a careless distribution of metal in the crank-case, as the guide pressure has still to be reckoned with and the pressure caused by tightening up the staybolts may be relied upon to cause serious distortion of a poorly ribbed case.

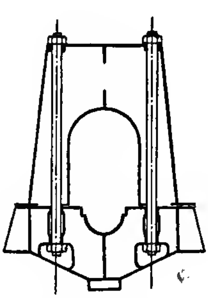


FIG. 80.

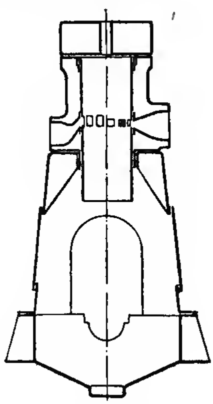


FIG. 81.

The cylinders being separate are secured either by a round, studded flange or by passing the staybolts through each corner of a deep, square flange of hollow section cast at the lower end

of the cylinder jacket for this purpose. The latter arrangement requires four staybolts for each cylinder, whereas with the former it is usual to arrange a pair of staybolts only at each main bearing girder. With the crank-case construction it is not necessary to make the side girders of the bedplate so strong as for an "A" frame type of engine, as the bending action referred to above is avoided and the bedplate and crank-case when bolted together form a girder construction of great rigidity. On the other hand, the upper part of the crank-case is clearly subject to bending actions similar to those which

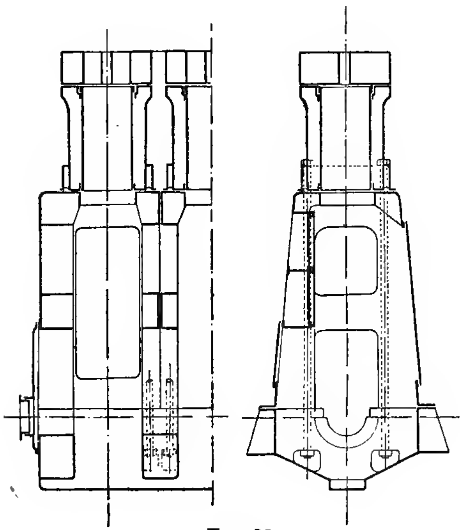


Fig. 82.

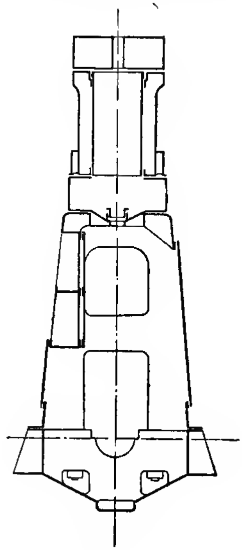


Fig. 83.

occur in the bedplate of an "A" frame engine and must be designed with this fact in view. Fig. 81 shews a section through a two stroke trunk engine of the crank-case type. Figs. 82 and 83 shew the crank-case construction applied to crosshead engines. The suitability of this type of framework for marine service has been amply proved in practice. In some cases the crank-case is common to two or more cylinders, and in others the case for each cylinder is a separate casting, the individual cases being bolted together to form a virtually continuous box of great strength and rigidity.

Trestle Type.—With this construction, illustrated in Fig. 84,

the cylinders are bolted to a base plate or entablature resting on trestle-shaped columns, the feet of the latter being secured to the main bearing girders. If the guide casting itself and its attachment to the trestles are sufficiently strong, this construction can compete with the crank-case type of frame in the matter of rigidity. The same effect could doubtless be achieved by some form of bracing. In Fig. 84 an alternative form of crosshead and guide is shewn, to which the trestle arrangement

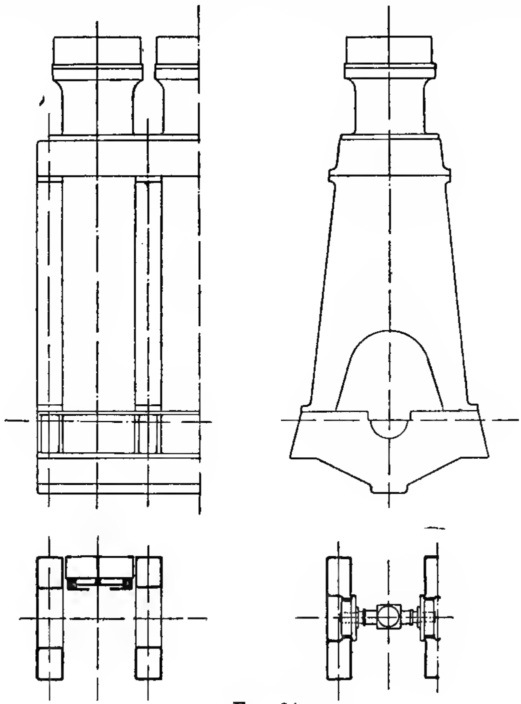


Fig. 84.

particularly lends itself, viz., the fore and aft double guide usually found on paddle steamers. The advantages of this form of guide for Diesel Engines are :—

1. Accessibility of running gear, the piston cooling gear in particular.
2. The guide blocks being free to adjust themselves to the guides are capable of sustaining a greater specific pressure, and consequently require less bearing surface than the usual type of guide shoe.

It is to be noted that the trestle construction involves some little extra care to retain lubricating oil when forced lubrication is used.

Staybolt Construction.—With this construction the cylinders are connected to the bedplate by means of turned bolts only, and the saving in weight on this account amounts to the considerable figure of about 25% of the complete weight of the engine. The reduction in cost of manufacture must also be considerable when the staybars are made of ordinary bright shafting screwed at the ends. Examples are shewn in Figs. 85

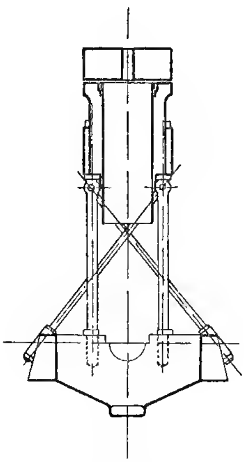


Fig. 85.

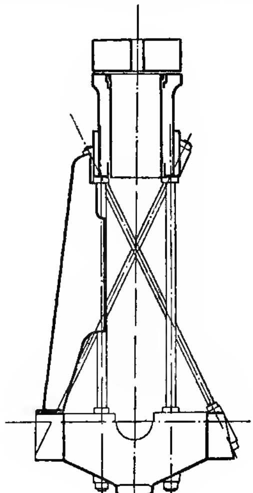


Fig. 86.

and 86 for trunk and crosshead engines respectively. In the latter the cast columns are relatively light, being designed for the guide pressure only, and their top ends are made free to slide, in order to avoid the subjection of the column to tensile strains on the compression and firing strokes. The difficulty of enclosing such engines adequately renders them unsuitable for forced lubrication under circumstances where economy of lubricating oil is a consideration; on the other hand, large slow-running engines, fitted with drip or ring lubrication, appear to be quite satisfactory when built on this principle.

Design of Bedplates.—The design of a suitable bedplate involves consideration of the following points, which will be dealt with in order, viz. :—

(1) The provision of a suitable main bearing.
(2) A girder construction under each main bearing, capable of supporting the full bearing load without central support.
(3) A sufficiently strong and stiff connection between the main bearing girders, forming at the same time an oil-tight tray.
(4) Suitable studding or staybolt arrangements for carrying the tensile pull of the columns.
(5) Arrangements for supporting the cam-shaft driving gear.
(6) Means for collecting drainage of lubricating oil to some convenient sump, whence it can readily be drawn off with a view to filtration and repeated use.
(7) Facings for barring gear, auxiliary pumps, etc.

Main Bearings.—Examination of badly worn crank-shafts indicates that the high bearing pressure obtaining for a short time when the piston is at its firing centre gives rise to far less abrasive action on the bearings than the less intense but longer sustained pressures due to inertia and centrifugal force in a four cycle engine. It appears that a film of oil is capable of sustaining a heavy pressure for a short time, but once the film has broken down, relatively feeble pressure is sufficient to cause abrasion, and there is small chance of the surfaces receiving a new film of lubricant until the pressure is removed. The result is that very little trouble is experienced with the lubrication of the main bearings of four stroke engines (the pressure on the journals being frequently reversed) unless the peripheral speed is so high as to reduce seriously the viscosity of the oil film by means of the heat generated by friction. In this case forced lubrication improves matters up to a certain point, but if the speed is still further augmented there comes a point when the bearings require to be water cooled.

With two cycle engines the direction of pressure is probably not reversed at all in most cases, and lubrication is consequently more difficult. When the peripheral speed is low, and the oil film in consequence as stable as possible, satisfactory results are obtainable even with ring lubrication of good design, if the maximum bearing pressure is kept about 30% lower than would be considered good ordinary practice with four stroke engines. If, on the other hand, high peripheral

speeds, or moderate bearing surfaces, or both, are required, then a system of high pressure forced lubrication would appear to be necessary, preferably in conjunction with a system of water cooling in extreme cases. The following table gives a rough guide to the limitations of the various systems of main bearing lubrication :—

System of Lubrication.	Peripheral Speed of Journal, feet per minute.	Projected Area of One Journal expressed as percentage of the Area of Piston.
FOUR STROKE ENGINES—		
Ring lubrication . . .	550	55%
Forced lubrication . . .	750	40%
Forced lubrication and water cooling	above 750	40%
TWO STROKE ENGINES—		
Ring lubrication . . .	550	75%
Forced lubrication . . .	700	60%
Forced lubrication and water cooling	above 700	60%

Drip or Syphon Lubricated Bearings are not used on land engines, but are sometimes fitted to Marine Diesel Engines of the open type. Apart from the fact that the caps have only to be proportioned to the inertia and centrifugal loads, these bearings are similar to those provided for steam engines, and need not be described here.

Ring Lubricated Main Bearings.—These are similar to the bearings fitted to electrical machinery and need not be described in detail. The arrangements for catching the oil squeezed out of the bearings and conveying it back to the oil well merit careful attention, as inefficiency in this direction leads to unnecessary waste of oil. In particular, the oil spaces and holes should be as large as possible, to avoid congestion. Fig. 87 shews a very usual form.

Forced Lubricated Bearings.—These follow high speed steam engine practice very closely, and the usual form of

staggering the circumferential oil groove in the top and bottom brasses is generally adopted. See Fig. 88.

Main Bearings Shells.—These are of cast iron in commercial work, and in the best practice the shells are tinned previous to the white metal being poured in. The chief essentials are :—

(1) Adequate thickness of shell.
(2) Good quality of white metal.
(3) Good adhesion between white metal and shell.

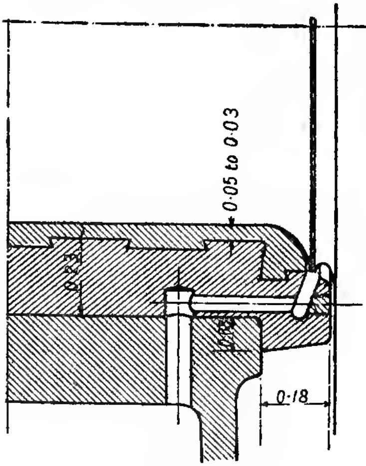


Fig. 87.

Common proportions are shewn in Figs. 87 and 88, the unit being the diameter of the journal.

The bearing cap should be designed as a beam capable of carrying a central load equivalent to the full inertia and centrifugal load due to one set of running gear. This is possibly a

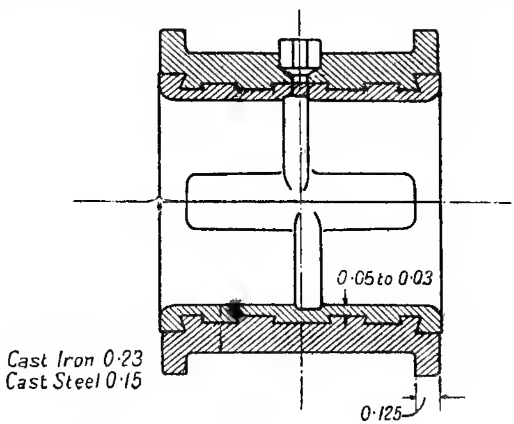


Fig. 88.

little on the safe side, but reference to Chapter V will shew that the margin is not large in the case investigated there.

Main Bearing Girder.—Where forced lubrication is used,

the main bearing girder may conveniently be of I section, the bottom flange being formed by the oil tray ; for ring lubrication a box section lends itself more conveniently to the formation of the oil reservoir. In any case the box section is preferable in the larger sizes. The depth of the girder is determined by that of the oil tray required to give an inch clearance or so to the connecting rod big end at the bottom of its path. Referring to Chapter V, it will be seen that the maximum reaction at a bearing for the case considered is equal to 0·8 of the resultant load due to pressure, inertia and centrifugal force, and this is the load for which the girder must be designed. In other cases the load may be less than this, but it is doubtful if in any case it approximates to the conventional load frequently assumed, viz., one-half the resultant cylinder load.

A very debatable point is the extent to which the oil tray can legitimately be regarded as a part of the tensile flange of the girder. The author's practice in this respect is to ignore the middle half of that part of the tray lying between two bearing girders (see Fig. 89). The span of the girder is the distance between the two points at which it meets the side girders.

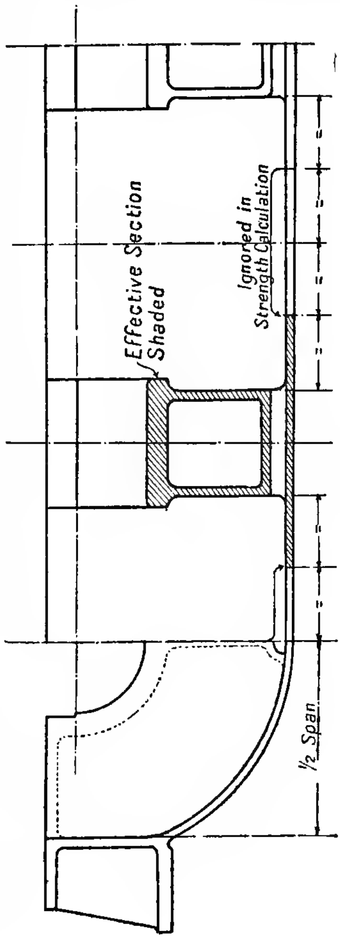


FIG. 89.

If W = Load on girder in lb.
l = Span in in.
M = Bending moment in in. lb.

Then $M = 0{\cdot}2\ Wl$ — approximately.

The assumption being that the fixing moments at the ends are negligible (which if not correct

is on the safe side) and that the load is distributed over the journal. Allowable stress 1500–2500 lb. per sq. in. for cast iron.

Side Girders.—With "A" frame engines the bending moment on each side girder may be taken as :—

$$\frac{\text{Half pressure load} \times \text{Distance between centres of bearings}}{6}$$

The usual stress allowance being about 1500 lb. per sq. in. Where the trestle or crank-case type of frame is used the side girders may be of lighter section (proportions will be given later).

Arrangements for Carrying Tensile Pull of Columns.—With the "A" frame construction the foot of each column is secured by a row of studs, the stress in which when referred to the normal maximum working pressure of 500 lb. per sq. in. in the cylinder amounts to about 5000 to 10,000 lb. per sq. in., according to the size of the stud. It is very convenient to have a list of the loads which studs and bolts of different sizes can conveniently carry, and such a list is given below :—

Size of Bolt or Stud (Whitworth).	Stress (Core) allowed, lb./in.^{2}	Working Load, lbs.
$\frac{1}{2}''$	2000	240
$\frac{5}{8}''$	2850	550
$\frac{3}{4}''$	3550	1080
$\frac{7}{8}''$	4250	1800
$1''$	5000	2750
$1\frac{1}{8}''$	5250	3650
$1\frac{1}{4}''$	5500	5000
$1\frac{3}{8}''$	6000	6300
$1\frac{1}{2}''$	7100	9300
$1\frac{3}{4}''$	8500	15,000
$2''$	9200	21,500
$2\frac{1}{4}''$	10,000	28,000
$2\frac{1}{2}''$	10,000	37,000
$2\frac{3}{4}''$	10,000	44,000
$3''$	10,000	54,500

Care must be taken that none of the studs are at any considerable distance from adequate supporting ribs. This is best

obtained by judicious spacing of the studs rather than the provision of special ribs for the purpose.

With the crank-case and trestle types staybolts are usually fitted, and in land work at any rate these should terminate within the bedplate and not penetrate to the under side of the latter for fear of oil leakage, which would destroy the concrete. The studs or bolts used to secure the crank-case to the

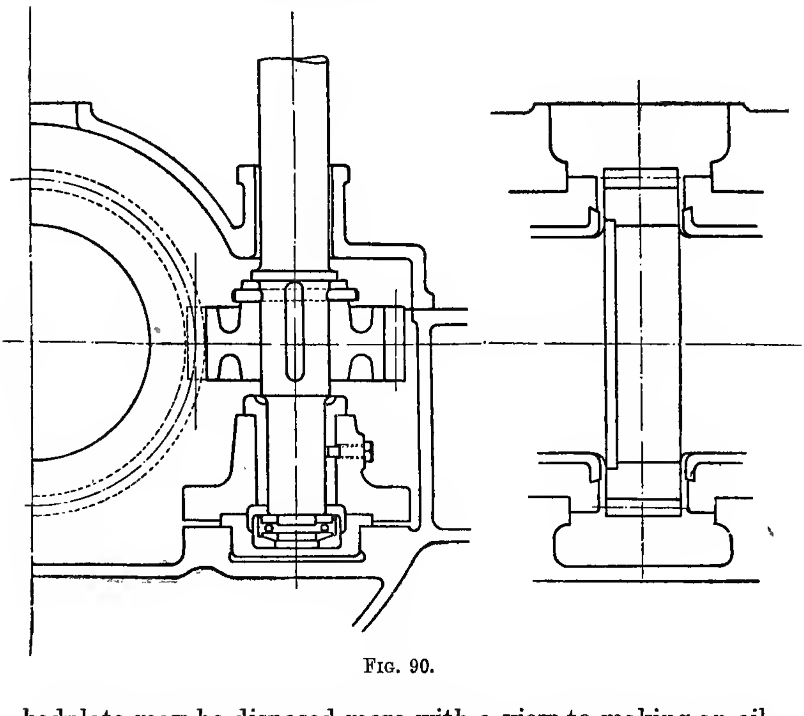


FIG. 90.

bedplate may be disposed more with a view to making an oil-tight joint than to carry any definite load. If staybolts are not fitted, then a sufficiency of effective bolt or stud area must be arranged in the neighbourhood of each column foot, and some of the bolts or studs must be inside the crank-case.

Cam-shaft Driving Gear.—The motion required by the valve gear is derived from the crank-shaft by spiral or spur gearing in the majority of designs. Fig. 90 shews a very common arrangement of spiral drive, with the driving-wheel between the two sections of a divided main bearing. It is good

practice to make the combined length of the two sections about 50% greater than the length of a normal bearing. There would appear to be nothing against having the spiral wheel outside the bearing altogether, provided the gear is at the fly-wheel end. This position for the valve gear drive is preferable to the compressor end as the weight of the fly-wheel tends to keep the journal in contact with its lower bearing shell, whereas the forward journal has freedom of motion (under the influence of forces which vary in direction) to the extent of the running clearance.

In six cylinder engines the spiral gear is frequently arranged at the centre of the engine, where it is very easily accommodated. There seems to be some feeling that the cam-shaft would whip unduly if driven from the end. This difficulty (if any difficulty can be said to exist) is easily overcome by making the cam-shaft about 10% larger in diameter than would be considered sufficient for a four cylinder engine.

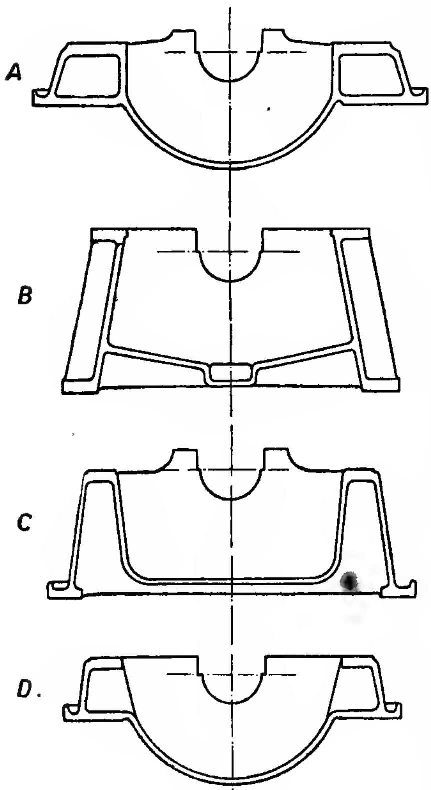


Fig. 91.

Where spur gearing is used for the valve gear drive, facings must be provided for the support of the first motion shaft.

Oil Drainage. — With land engines of the non-forced lubricated type the oil which drips down from the cylinders and is thrown from the big ends is drained periodically from the forward end of the bedplate and holes are cored through the main bearings girders to give the oil free passage. Perhaps the best arrangement is a rectangular duct about four inches square running down the centre of

the oil tray. Small holes are useless as they are easily choked. With forced lubricated engines the same arrangements are made with the addition of a collecting sump of good capacity, a pump for forcing the oil into the bearings and filters in duplicate. These features being familiar in steam engine practice need not be described in detail. It must be borne in mind, however, that where trunk engines are being considered the oil is contaminated with carbon, so that the filtering arrangements require to be on a more liberal scale than is necessary with engines in which the cylinder is isolated from the crank-case.

Proportions of Bedplate Sections.—Fig. 91 gives approximate proportions for various types of bedplate sections, the unit being the stroke of the engine. Type "A" is usually associated with the "A" frame construction. Type "B" is a useful one for main or auxiliary marine engines as it enables the engine to be bolted direct to a tank top or to a deck without building up a special seating.

Type "C" is preferable to type "A" for land generating sets as the extra depth of bedplate enables the generator to be flush with the engine room floor without the necessity of building the engine on an unsightly plinth. A deep bedplate is also very desirable with six cylinder engines as the cancellation at the centre of the engine of the inertia and centrifugal couples gives rise to vibrations, the amplitudes of which are reduced by increasing the stiffness of the framework.

The general thickness of metal may be about 4% of the cylinder bore increased to about 6% or 8% on machined surfaces. These figures are usually exceeded on small engines on account of the difficulty of obtaining consistent results in the foundry with thin metal. The following figures for different diameters of cylinder represent good practice :—

Bore of Cylinder, in.	General thickness of Metal for Bedplate, in.
10	$\frac{5}{8}''$
12	$\frac{3}{4}''$
15	$\frac{13}{16}''$
18	$\frac{15}{16}''$
21	$1\frac{1}{16}''$
24	$1\frac{3}{16}''$
27	$1\frac{1}{4}''$
30	$1\frac{3}{8}''$

The above are useful as a guide, but must not be relied upon without check calculations, as special constructions may require local strengthening to keep the stresses within a safe figure.

"A" Frames.—An "A" frame for a four stroke trunk engine is shewn in Fig. 92; the discussion of those parts which are common to separate cylinders as distinguished from cylinders combined with columns will be reserved for a separate chapter. The liner is a push fit or light shrink fit in the upper flange of the jacket. The fit at M should be a few thousandths slack, to prevent seizure at this point. The jacket is swelled locally, to give adequate water passage, and six or eight strong ribs are provided to transmit the tensile load across this section, which would otherwise be greatly weakened by the discontinuity of the wall. At Q the liner is a push fit, allowing the liner to expand axially relatively to the jacket. Sometimes a stuffing-box is fitted to prevent water leakage. P is the water inlet connection. L L are bosses to accommodate lubricator fittings. N is a cleaning door.

Strength of "A" Frames.—Fig. 92 is drawn for a 15 in. cylinder, the stroke being 21 in., and the dimensions given represent good average practice. The stresses may be checked as follows:—

Maximum working pressure . . . 500 lb. per sq. in.

" " " load $0{\cdot}784 \times 15^2 \times 500 = 88{,}000$ lb.

Tensile stress in parallel part of jacket (mean dia. $= 23{\cdot}5''$)

$$= \frac{88{,}000}{\pi \times 23{\cdot}5 \times 1} = 1190 \text{ lb./in.}^2$$

Next consider section "AA" of one leg. For this section conditions are worst if the nuts at the foot are not tight and the reaction at the foot consists of a simple vertical pull of 44,000 lb. Referring to Fig. 92, the direct tensile pull in the leg is 42,000 lb., and the sectional area at "AA" being 57 sq. inches,

Direct tensile stress at "AA"		$=42{,}000 \div 57 = 737$ lb. per sq. in.
Shear stress	"	$=13{,}500 \div 57 = 237$ " " "
Bending moment	"	$=44{,}000 \times 9{\cdot}5 = 42{,}000$ in. lb.
Section modulus	"	$= \sim 230$ in.^{3}
Bending stress	"	$=420{,}000 \div 230 = 1825$ lb. per sq. in.
Total tensile stress	"	$= 1825 + 737 = 2562$ " " "

This stress refers to the outside fibres of the column, and is probably in excess of what actually occurs as the fixation of the foot by the holding down studs produces a moment in the reverse direction.

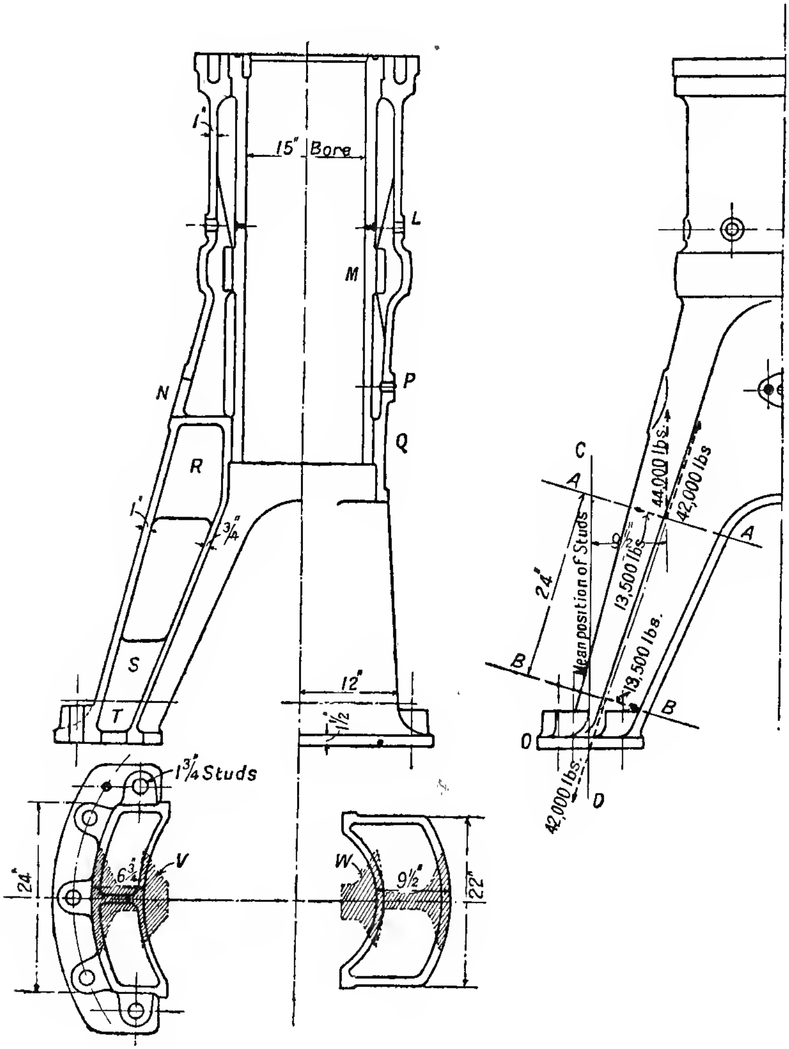


FIG. 92.

Section " BB " is subject to maximum stressing action if the foot is securely fixed to the bedplate, as it should be. The vertical reaction of 44,000 lb. is again resolved into a direct pull along the axis of the leg amounting to 42,000 lb., and a shear of 13,500 lb. The area " BB " being 64 in.^{2}, therefore the direct tensile stress at " BB " $=42,000 \div 64 = 657$ lb./in.^{2}

The shear load of 13,500 lb. at " BB " being opposed by an equal but opposite shear load at " AA " there must be a couple of magnitude $13,500 \times 24$ in. lb. to keep the part of the leg lying between " AA " and " BB " in equilibrium. Assuming that this couple is composed of two equal parts operating at " AA " and " BB,"

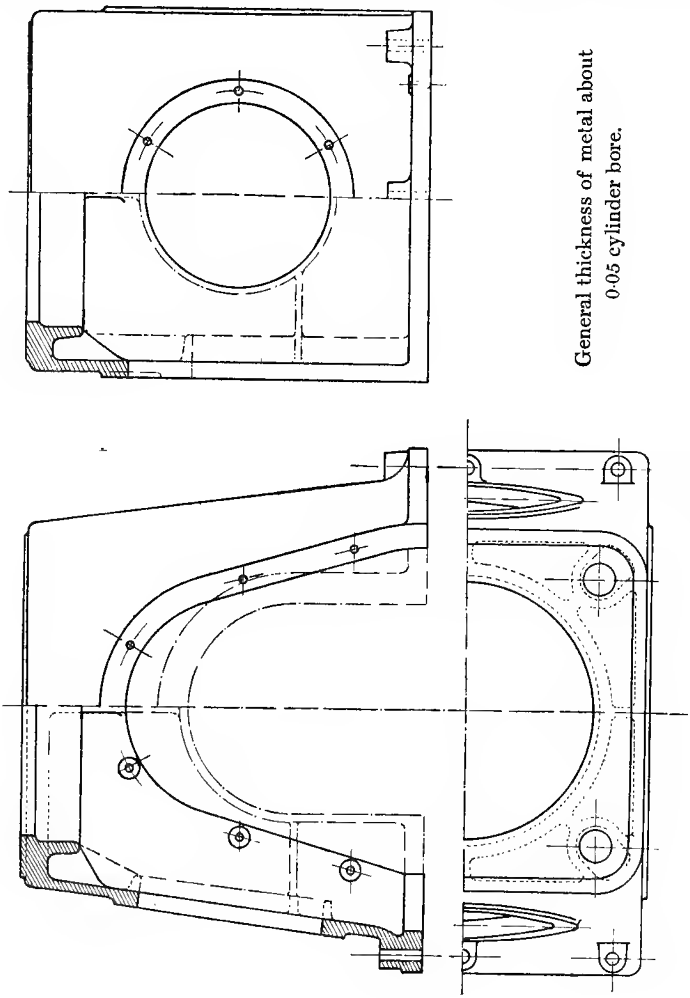
Bending moment at "BB"		$=13,500 \times 24 \div 2 = 162,000$ in. lb.
Section modulus	,,	$= \sim 170$ in.^{3}
Bending stress	,,	$=162,000 \div 170 = 953$ lb. per sq. in.
Total tensile stress .	.	$= 953 + 657 = 1610$,, ,, ,,

The stress in the studs depends upon the degree to which the nuts are tightened, and if of sufficient area the stress is probably not affected by the applied load. It therefore only remains to see if the loads induced by tightening them up to their nominal working stress are sufficient to prevent the joint opening.

Referring to the table on page 130, the nominal load of five $1\frac{3}{4}$ in. studs $=75,000$ lb. Subtracting 44,000 lb. there remain 31,000 lb. to resist tilting about the outside edge 0 of the foot. The distance from 0 to the centre of mean position of the studs is $6\frac{1}{2}$ in., and the corresponding moment is therefore $31,000 \times 6{\cdot}5 = 202,000$ in. lb., which is greater than the moment to be resisted.

Design of Crank-cases.—A simple form of crank-case is shewn on Fig. 93, stroke to bore ratio 1·25. The crank-case is machined on each side and in position is bolted to its neighbours, an arrangement which, though unusual, has its advantages, both in the factory and outside, as the small sections are easier to handle than a crank-case in one piece. Provision is made for staybolts, and the thicknesses of metal shewn are about the minimum to give satisfactory results with this design. It must not be thought that these proportional thicknesses are capable of substantial reduction with large sizes of engine without modification to the distribution of metal. On the other hand, if the interior of the case is built up in girder or box formation,

or generally reinforced by internal ribbing, as shewn dotted in Fig. 93, the general thickness may be reduced by about 25%, and advantage is taken of this fact in designing large engines, the castings of which would be of undesirable thickness if the simpler forms were adopted. In the event of staybolts not being used, it is desirable to check the stresses in the legs in the manner described for an " A " frame. Judging by successful



General thickness of metal about 0·05 cylinder bore.

Fig. 93.

designs, the use or omission of the staybolts has little influence on the strength of crank-case required, and this is readily explained by the fact that whereas the staybolts relieve the crank-case of tensile stresses, the tightening of the former throws a heavy buckling load on the crank-case, perhaps double the tensile load due to the working pressure in the cylinders. These considerations do not apply, however, to those designs in which the staybolts are extended upwards to the cylinder cover. In these cases the crank-case has only the guide pressure to contend with. On the other hand, the use of

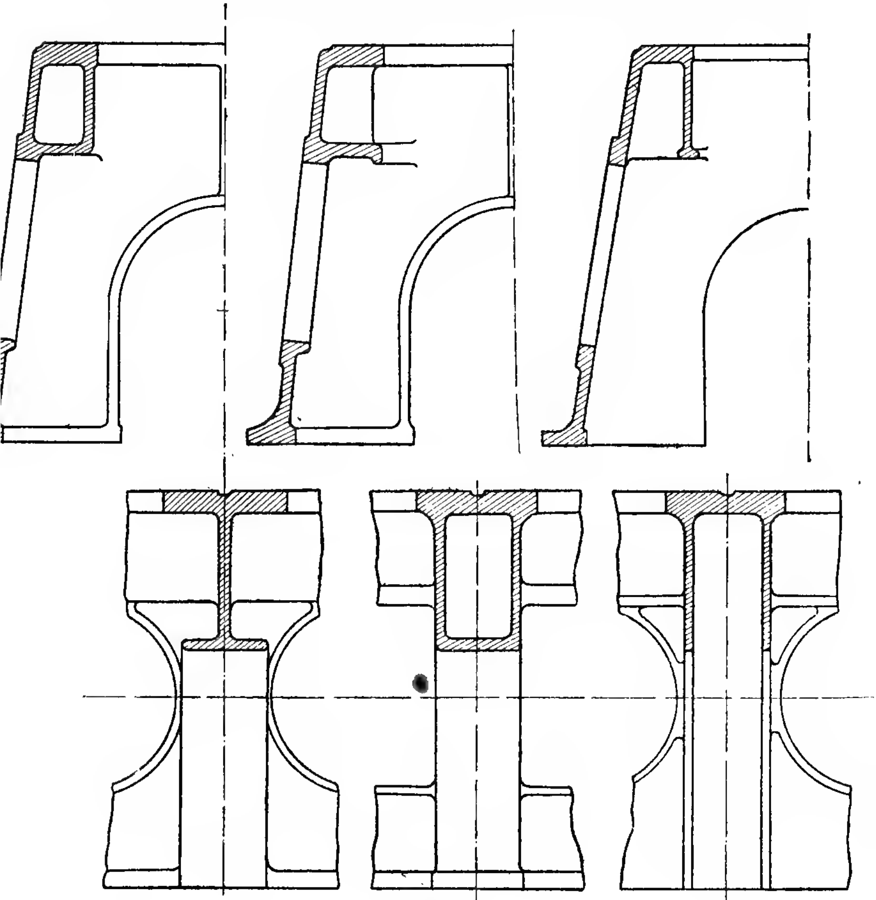


FIG. 94.

staybolts passing through lugs on the cylinders enables the latter to be pitched closer together than would be easily possible otherwise, and in any case they add strength and rigidity at very little expense and increase in weight.

The above notes on the strength of crank-cases, as well as the figures for the thickness of metal, apply equally to cross-head as to trunk engines. The additional height of the former has little if any influence on the matter, as the guide reaction acts at the same height, above the centre line of the crank-shaft, assuming the same length of connecting rod in each case. Some alternative forms of construction are shewn in Fig. 94. In the case of very small engines the use of the minimum thickness of metal allowable on considerations of strength and rigidity only, would give rise to trouble in the average foundry and additional thickness is usually given. In the following table foundry considerations are neglected, as these must be dealt with by the designer in each individual case, in accordance with his judgment of the capabilities of the foundry in question, and in this matter the foundry manager will be able to give assistance.

Bore of Cylinder, in.	General thickness of Crank-case metal, in. Plain type, Fig. 93.	General thickness of Crank-case metal, in. Box or girder formation, Fig. 94.	Diameter of Whitworth Staybolts, in.
10	$\frac{1}{2}''$	$\frac{7}{16}''$	$1\frac{5}{8}''$
12	$\frac{5}{8}''$	$\frac{1}{2}''$	$1\frac{3}{4}''$
15	$\frac{3}{4}''$	$\frac{9}{16}''$	$2\frac{1}{8}''$
18	$\frac{7}{8}''$	$\frac{3}{4}''$	$2\frac{1}{2}''$
21	$1\frac{1}{16}''$	$\frac{7}{8}''$	$2\frac{3}{4}''$
24	$1\frac{3}{16}''$	$1''$	$3\frac{1}{8}''$
27	$1\frac{3}{8}''$	$1\frac{1}{16}''$	$3\frac{3}{8}''$
30	$1\frac{1}{2}''$	$1\frac{3}{16}''$	$3\frac{3}{4}''$

The figures for the diameters of the staybolts are based on the assumption that they carry the whole pressure load. In cases where the cylinders are secured to the crank-case by a studded flange the staybolts if fitted at all may be made considerably lighter, according to judgment or the results of experiment. Other points to be considered in designing a crank-case are :—

(1) The provision of oil-tight access doors of ample size for overhauling the bottom ends.
(2) End casings provided with oil flingers, stuffing boxes, or other means of preventing the escape of oil.
(3) Facings, and other necessary accommodation for valve gear.
(4) Bosses to carry lubrication oil connections to the main bearings.
(5) Facings for platform brackets.
(6) A vent pipe or valve of large area, to relieve pressure in the event of an explosion in the crank-case without loss of lubricating oil during normal working.
(7) Steady pins to each section of the case, to fix correct location.

Machining the Framework generally.—In designing all parts of an engine the designer will keep in mind the capabilities and limitations of the manufacturing plant and the operatives. This is especially necessary in the case of the framework, on account of the relatively large size of the parts. Where the most modern type of face milling plant is available the element of size offers no difficulties, and bedplates of 60 feet in length may be faced in one operation. Where planing must be resorted to the capacity of the machines must be studied in the early stages of the design. Machined faces should be arranged in as few different planes as possible, and ribs or flanges projecting beyond those planes are to be avoided as much for convenience in machining as for the sake of appearances. The simpler forms of girder or box-girder construction are to be preferred to those designs in which alternate perforation by lightening holes and reinforcement by ribbing mutually defeat each other's object. The lightest, strongest and cheapest forms are to be attained with a minimum of holes and ribs when cast iron is used. Large steel castings, however, are preferably lightened out almost to the extent of lattice-work, in order to facilitate rapid stripping of the cores after solidification and to minimise initial stresses.

Literature.—The different types of framework used in Marine Diesel Engine construction, and the forces acting on them, are discussed in the following paper :—Richardson, J., "The Development of High Power Marine Diesel Engines," Junior I.E., April 20th, 1914.

CHAPTER VIII

CYLINDERS AND COVERS

General Types.—The great majority of Diesel Engines are provided with cylinder liner, jacket and cover as separate pieces, as in Fig. 95, which refers to a four cycle trunk engine. Different arrangements have, however, been used successfully, and deserve mention. With small engines, simplification is achieved by casting the jacket and liner in one piece, as in Fig. 96. Remembering that the bulk of the jacket wall remains

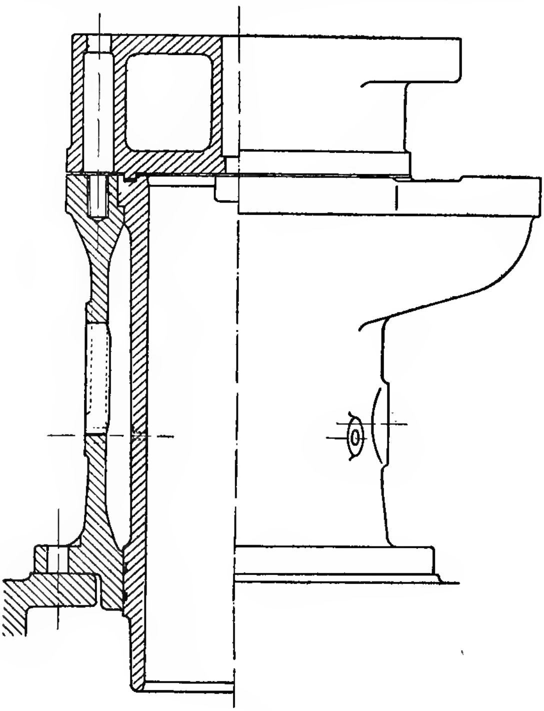


Fig. 95.

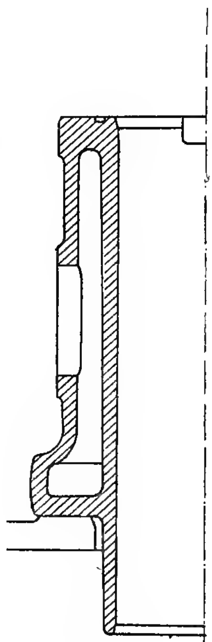


Fig. 96.

stone cold, it will be appreciated that this construction involves increased tensile stresses on the jacket, due to the tendency of the liner to expand, and jackets of this kind have been known to crack circumferentially. In cases where staybolts have been fitted to carry the tensile stresses from the cover downwards little damage has resulted. On the other hand, when the jacket has been relied on for this function, rupture during working may easily occur, and has sometimes resulted in the cylinder being projected towards the roof. These considerations would appear to indicate that the use of this construction, without staybolts or other safeguards, is not lightly to be attempted without serious consideration of the capabilities of the foundry. In Fig. 97 is shewn a construction in which the cover is incorporated with the cylinder casting in motor-car style. In this case the tensile stresses are mainly carried by the liner, and the jacket is made relatively thin and flexible. This design, though probably safer than that of Fig. 96, also makes some demand on the skill and care of the foundry people. In this connection it is worth while bearing in mind that many failures might possibly have been avoided if it had been realised that certain continental designs, in which lightness has been a primary consideration, were only practicable if the greatest care were exercised in the selection of material and in making the castings. There are other types of cylinder in successful use, notably those in which the liner and cover are cast together apart from the jacket, but the remainder of this chapter will be devoted to the consideration of the details of the more common construction, in which the liner jacket and cover are separate pieces. Unless the contrary is stated, cast iron is understood to be the material in each case.

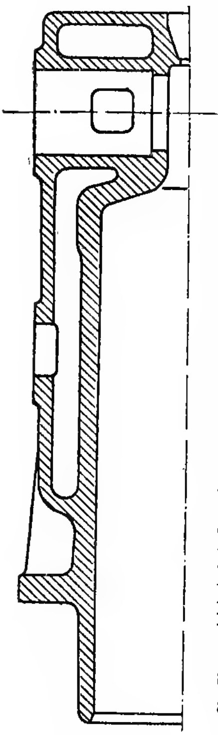


Fig. 97.

Liners.—Special cast iron is used for liners, but there is little unanimity of opinion as to the most desirable properties beyond the obvious requirements of soundness and homogeneity. The greatest difficulty to be overcome is abrasion by the piston rings. At present it seems open to question whether the

problem is most influenced by the material of the liner or the piston rings themselves. It is very seldom that liners crack* either in four or two stroke engines, and this immunity is doubtless due to the rapid conduction of heat from the breech end to those parts which only come in contact with gases at relatively low temperatures, from which it would appear that the best way to cool a relatively inaccessible spot is to connect

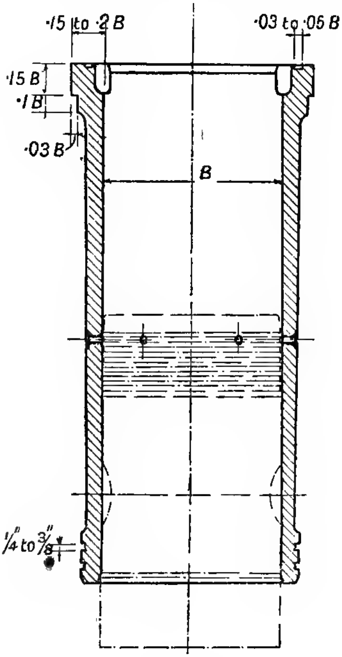


FIG. 98.

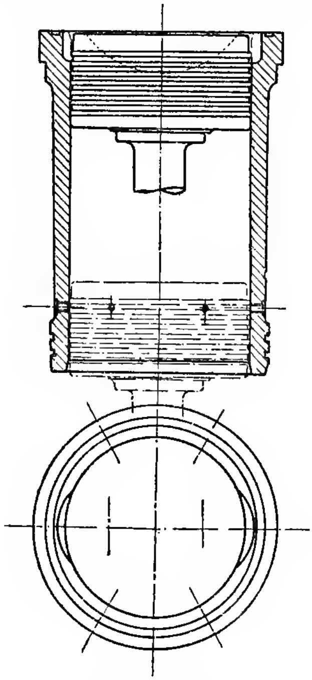


FIG. 99.

the latter to a large, well-cooled area, as near to it as possible. This principle will be referred to again in connection with exhaust valve casings.

Typical liners for four stroke engines of the trunk and crosshead types respectively are shewn in Figs. 98 and 99. With the latter the piston is only of sufficient length to carry the rings, and the length of the liner is determined by the position of the bottom ring at the bottom of the stroke. With the trunk engine the liner must be long enough to embrace a sufficient

* Except at a piston seizure.

length (about equal to the bore) of the parallel part of the piston when at the bottom of its stroke, in order to avoid a piston knock at the bottom dead centre. It is therefore necessary to determine the clearance volume and complete the design of the piston before the length of the liner can be fixed finally.

The bore is usually parallel with four stroke engines and slightly barrelled in way of the ports in two stroke engines to allow for the restraints which are inevitably placed at that position against free expansion of the liner. Probably the best bore is produced by finishing with a reamer in a vertical machine. Grinding is frequently adopted, but there is a question if this process does not to some extent destroy those properties of cast iron which facilitate good lubrication. The outside surface is frequently left unmachined in competitive work, and there is probably no serious objection to this practice for four cycle work. For two cycle engines it seems reasonable to take advantage of the increased heat conductivity obtainable by removing the skin.

Strength of Liners.—The upper end of the liner is subject to a working pressure of about 500 lb. per sq. in., and the thickness at this part measured under the heavy top flange may be found by the following formula, which represents average practice for substantial engines :—

$$\text{Thickness} = 0{\cdot}08 \text{ bore} + \tfrac{1}{8}''$$

The working stress being about 3000 lb. per sq. in. in the case of a 30 in. cylinder, and less in smaller sizes ; explosions at starting, etc., may nearly double this stress occasionally. Unfortunately the available information on the effects of repeated stress is not sufficiently complete at present to enable one to say definitely whether or not these excesses of stress have any influence on ultimate failure by fatigue, but the writer is inclined to believe (on the strength of such evidence as has come before his notice) that the elimination of these occasional excess pressures would not enable any substantial reduction of thickness to be effected with the same margin of safety.

On account of the diminution of pressure on expansion the liner may be tapered to a thickness of about 0·04 bore at the open end.

The breech end of the liner requires to be reinforced by a

heavy flange, to avoid distortion due to the pressure of the cover on the spigot joint. Proportions are given in Fig. 98.

Points of Detail.—The difficulty of accommodating the valves in the limited space available in the cover of a four stroke engine usually renders it necessary to make recesses in the top of the liner to clear the air and exhaust valve heads (see Fig. 98). Four or more tapped holes are provided in a circumferential line round the liner to accommodate the lubricating fittings, these being drilled when the liner is in position in the jacket. The holes are located at about the level of the second piston ring (counting from the top), when the piston is at the bottom dead centre. The fittings themselves will be described later. The water-joint between liner and jacket at the lower end may be made by one or more rubber rings. The joint between cover and liner may be of copper or asbestos compositions.

Two stroke liners are complicated by exhaust, and sometimes air ports (see Fig. 102). In the earlier designs the bars between the latter were always provided with water passages, which introduced difficulties in manufacture, and the value of which seems doubtful, and these are now frequently omitted. The fitting surfaces at this point are preferably ground, to minimise chance of leakage, as the high temperature prohibits the use of rubber packing rings.

Cylinder Jackets.—A simple and effective form of jacket for a four cycle engine is shewn in Fig. 95, and in this example the jacket takes the pressure pull without the assistance of staybolts. The chief points to be observed are :—

(1) A heavy flange at the top to carry the liner and to enable the tensile forces concentrated at the studs to distribute themselves uniformly round the jacket without producing high local stresses.

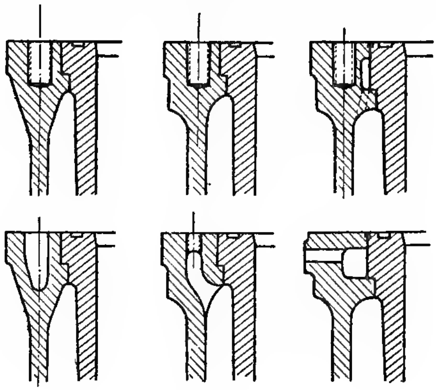
(2) A nearly plain cylindrical barrel, as nearly as possible in line with the pitch circle of the cover studs and provided with sludge doors, bosses for lubricating fittings, and a bracket for supporting the cam-shaft bearings.

(3) A circular flange at the bottom for securing to the crankcase.

The remarks *re* tensile forces under heading (1) apply here also, but to a less degree, as the studs are pitched closer together than would be feasible on the cover. On these consider-

ations the thickness of the jacket for equal strength should taper gently towards the middle, and the form shewn in the figure is the practical compromise. Some of these points will be considered in greater detail.

Top Flange of Cylinder Jacket.—In small engines this may be solid, but with larger sizes, say from 15 in. bore and upwards, difficulty is sometimes experienced in obtaining sound metal at this point, and coring of the flange between the studs is resorted to in order to accelerate cooling in the mould. Different constructions are shewn in Fig. 100. Schemes A and



C. B. A.

Fig. 100.

B have the additional advantage of increasing the cooling surface. Where four cycle engines are concerned the importance of this consideration is probably negligible. Scheme B requires a water outlet connection between each stud if air pockets are to be avoided. On the other hand, the expense of coring is less than with scheme C.

Barrel of Cylinder Jacket.—This is sometimes conical, instead of cylindrical, and in this case it is reasonable to provide a vertical internal rib under each stud to discount the additional stress involved. Consideration of manufacturing costs, and of the good appearance of the engine, rule out of court any form of external ribbing. The brackets supporting the valve gear take many forms in different designs. That shewn in Fig. 95 is the modern form, and considered in conjunction with the gear it supports appears to combine most advantages, including that of elegance.

Bottom Flange of Jacket.—If four staybolts are provided for each cylinder, these may conveniently be used to secure the latter to the crank-case. The concentration of the tensile load at four points necessitates a heavy flange, preferably of box form, as shewn in Fig. 101. The corners of this flange being each subject to a load of one-quarter of the maximum working pressure load, deserve attention in the form of a calculation of the bending stress involved. A plain, square shape would appear to be preferable to some of the more elaborate shapes which have occasionally been used, the flat sides lending themselves well to the provision of facings for various purposes.

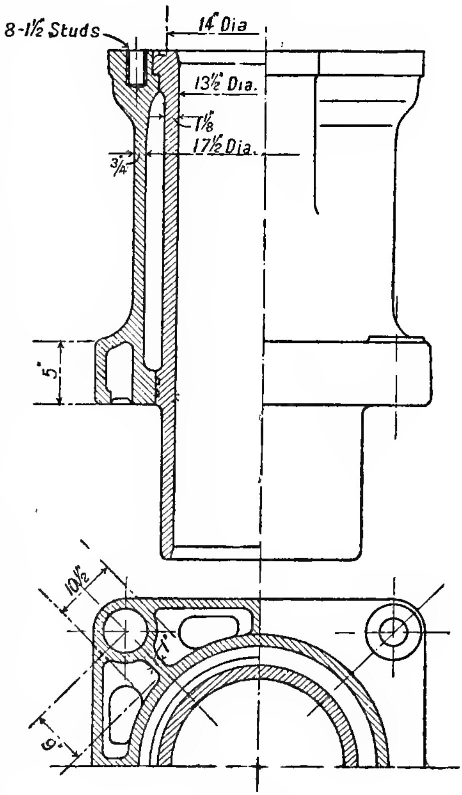


Fig. 101.

Frequently the flange is spigoted into the top of the crank-case, but as this involves an unnecessary machining operation on the latter and makes cylinder alignment more difficult, the better practice is to core the aperture in the crank-case sufficiently large to allow for adjustment of the position of the cylinder and to locate the latter by means of two steady pins.

Strength of Four Stroke Cylinder Jackets.—The considerations of strength which enter into the design of a cylinder jacket are illustrated by the following check calculations relating to the cylinder shewn in Fig. 101.

Bursting stress in liner

$$=\frac{500 \text{ (lb. per sq. in.)} \times 6{\cdot}75}{1{\cdot}125}=3000 \text{ lb. per sq. in.}$$

Nominal pull in each cover stud $=\dfrac{500\times 0{\cdot}784\times 14^2}{8}=9650$ lb.

Permissible nominal load for $1\frac{1}{2}$ in. stud, according to table on page 130, 9300 lb.

Maximum working pull in jacket $=0{\cdot}784\times 13{\cdot}5^2\times 500=$ 71,500 lb.

Tensile stress in jacket $=\dfrac{71{,}500}{\pi\times 18{\cdot}25\times 0{\cdot}75}=1670$ lb. per sq. in.

Owing to the peculiar shape of the bottom flange the calculation of its strength presents a difficulty which is easily evaded by substituting for the actual section a simpler one of obviously inferior strength.

Nominal load at each corner, 71,500 lb. $\div 4=\sim 18{,}000$ lb.

Moment from centre of bolt to jacket wall $=18{,}000\times 9$ in. lb.

Modulus of hypothetical section :—

$$z=\left(\frac{10{\cdot}5\times 5^3}{12}-\frac{9{\cdot}5\times 3{\cdot}5^3}{12}\right)\div 2{\cdot}5=30{\cdot}2\text{ in.}^3$$

$$\therefore\ \text{Stress}<\frac{18{,}000\times 9}{30{\cdot}2}\quad \text{i.e.}\ <5{,}400\text{ lb. per sq. in.}$$

In view of the unfavourable assumptions this is probably not excessive for first-class cast iron.

Jackets for Two Stroke Engines.—The necessity for providing exhaust passages or belts, and in some cases passages for scavenge air as well, introduces considerable complication into the design, renders the stresses in certain parts more or less indeterminate, and makes greater demand on the skill of the manufacturing departments, in comparison with that required by four cycle construction.

Referring to Fig. 102, it will be seen that the exhaust belt interrupts the vertical line of the jacket wall, and if the latter has to carry the main tensile stresses internal ribbing becomes a necessity. The arrangement shewn is perhaps as good as any, but the attachment of ribs to the exhaust belt has a restraining influence on the temperature expansion of the latter which can only result in mutual stresses. It appears, however, that these are not very serious, as cylinders which have failed in other respects have remained intact at this point. Fig. 103 shews a construction in which a good attempt is made to secure continuity of the vertical wall of the jacket. Either of these systems is probably satisfactory for cylinders of medium size. Large cylinders, however (and this applies to

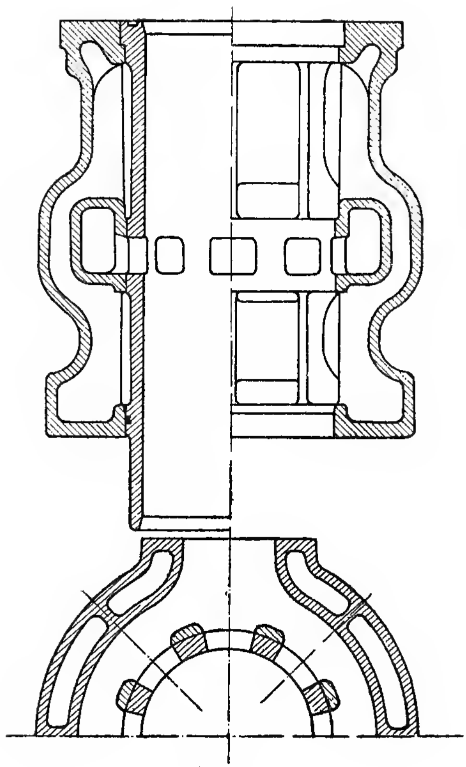


Fig. 102.

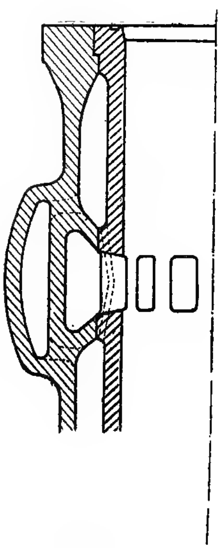


Fig. 103.

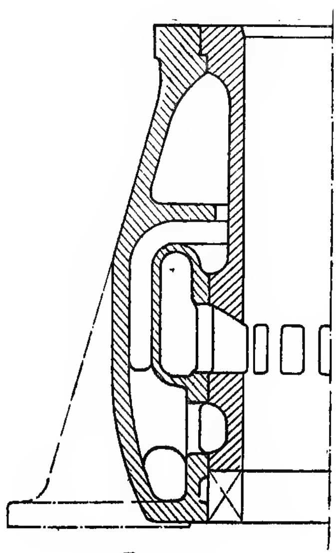


Fig. 104.

other parts as well), are known to be subject to greater temperature differences than smaller ones (though not to the extent sometimes suggested), and the leading designers have had recourse to other expedients when faced with the problem of constructing cylinders of large size.

In Fig. 104 the jacket wall may be described as similar to a honey-pot in shape and of abnormal thickness, to allow for the bending stresses caused by the curvature of the walls and the fact that the tensile supporting forces are localised at two feet.

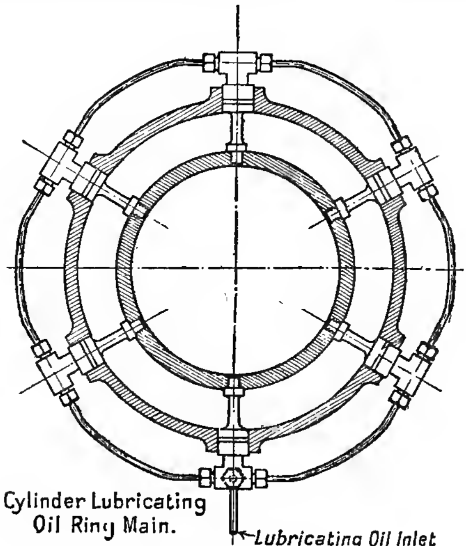


FIG. 105 (See page 152).

The exhaust belt is of relatively thin metal, with comparatively small support from the walls. It will be evident that the strength of the jacket is very slightly influenced by the exhaust belt, and that the latter is free of all but temperature stresses. This construction, therefore, attains a good approximation to the correct allocation of the respective duties of jacket and exhaust belt.

As disadvantages, may be cited abnormal weight of cylinder and the difficulty of casting a cylinder involving widely different thicknesses of metal.

Another and perhaps better way out of the difficulty is to connect the cylinder cover to the bedplate by means of stay-

bolts, thus relieving the jacket of all stresses except those induced by temperature differences. The jacket in this case virtually hangs from the cylinder cover, and only requires to be attached thereto by studs proportioned to a load based on the cylinder pressure and the annular area lying between the cylinder bore and the spigot at which the cover joint is made. The upper flange is preferably made fairly substantial, but other thicknesses may be made a practical minimum.

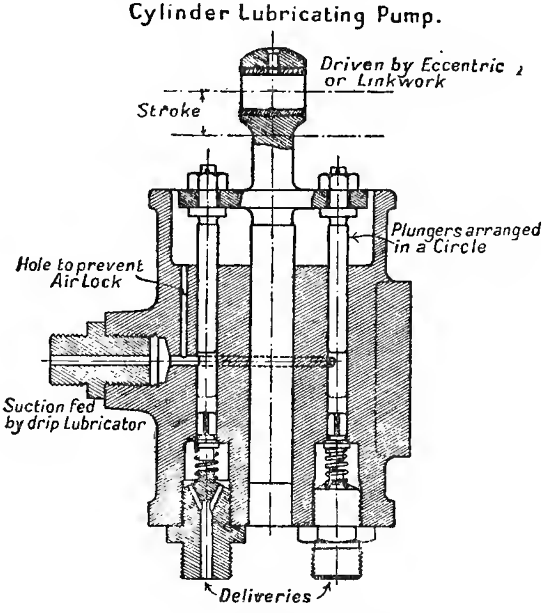


FIG. 105 (See page 152)

Cylinder Lubrication.—The problem of cylinder lubrication in Diesel Engines consists in effecting uniform distribution of minute quantities of oil. The quantity of oil admitted must be the minimum necessary to effect satisfactory lubrication, as the oil " cracks " in service, leaving a gummy deposit, which in course of time causes the piston rings to stick. Under favourable conditions this may be several months, even a year. Every drop of superfluous oil reduces this period, hence the importance of uniform distribution so that every part may have sufficient, but none a superfluity. These conditions are

best secured by a separate controllable feed to each of about six or eight points round the circumference of the cylinder. A typical lubricating fitting is shewn in Fig. 105 (pages 150–152), and the point to be observed is that the fitting must adapt itself to slight relative movement between the liner and jacket. The small hole at the end which leads to the surface of the liner reduces to a minimum the chances of the fitting becoming choked with carbon. With forced lubricated engines, in which the cylinder is not isolated from the crankpit, it frequently happens that more than sufficient oil reaches the cylinder, apart from any arrangements made for the purpose. In this case the problem may be to devise scraper rings, vent holes, or other devices, to remove the superfluous oil.

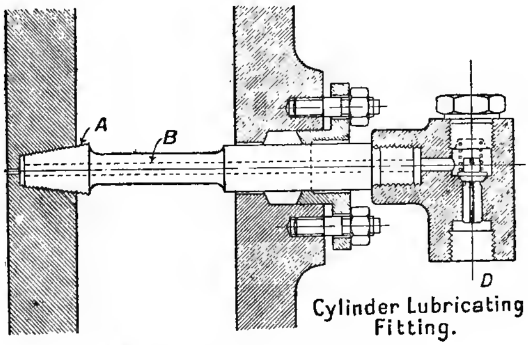


FIG. 105 (See also pages 150, 151).

Cylinder Covers.—Owing to a considerable number of failures in service and difficulties experienced in manufacture, cylinder covers for both four and two cycle Diesel Engines have come to be regarded as difficult pieces of design, and it may perhaps be instructive to review the subject in a more or less historical manner.

The earlier type of four cycle cover is shewn in Fig. 106, from which it will be seen that the internal coring is complicated and that a few core-holes of small diameter only are provided to vent the core in the mould. In spite of these disadvantages, such covers have given good service when made by the most skilful of continental manufacturers. Dismissing for the moment the question of manufacturing costs, these covers have the following shortcomings :—

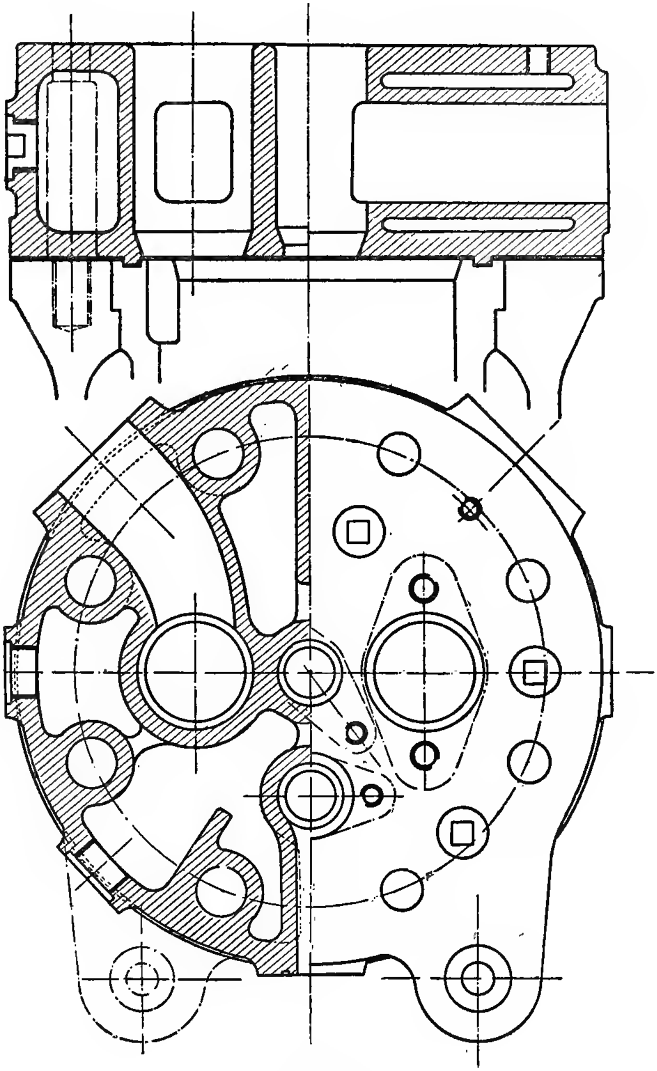


FIG. 106.

(1) The thin walls of uncooled metal between the recesses for fuel and exhaust valves are liable to crack on overloaded engines.
(2) The hot exhaust passage is too rigid to permit of much expansion and leads to cracking of the bottom plate.
(3) The small core-holes give poor access to the interior for purposes of cleaning away accumulated scale.

Assuming first-class foundry work, the two latter considerations are perhaps the most important. Modern development is on the following lines :—

(1) The provision of large doors, which serve the double purpose of providing good access for cleaning and affording better support and venting for the core when casting.
(2) Elimination of all internal ribs, as experience seems to shew that the tubular walls provided to accommodate the valve casings provide all requisite support between the top and bottom plates.
(3) Using brass or steel tubes expanded into the recesses for the fuel valve and holding-down studs.
(4) The use of square instead of conical seats for the valve casings.

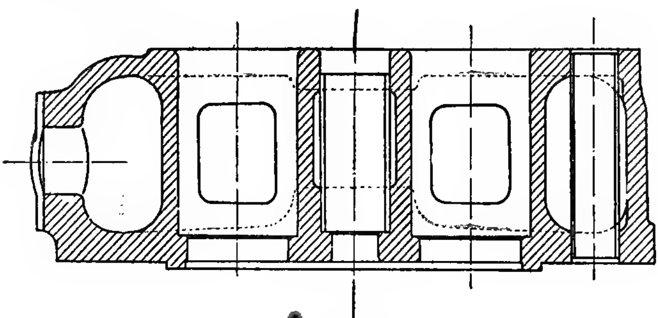


FIG. 107.

A cover designed on these lines is shewn in Fig. 107. Some makers have simplified the question of casting at the expense of introducing extra machining and fitting operations by making the top plate a separate piece (see Fig. 108.)

Another innovation which is becoming increasingly common is to place the fuel valve off centre. This arrangement enables

the cooling space around the fuel valve to be increased, but too great a displacement of the fuel valve from the centre position necessitates a special shape of combustion space.

Points of Detail.—Owing to the large recesses for the valve cages, a four cycle cover is relatively weak, considering the amount of metal in it, and on this account all stud holes should be well bossed under and all inspection openings well reinforced by compensating rings like a boiler.

The under face of the cover is machined all over, but on the top face machining is sometimes restricted to those parts which are occupied by valves, etc. This enables the corners to be given a liberal radius, which in addition to improving the

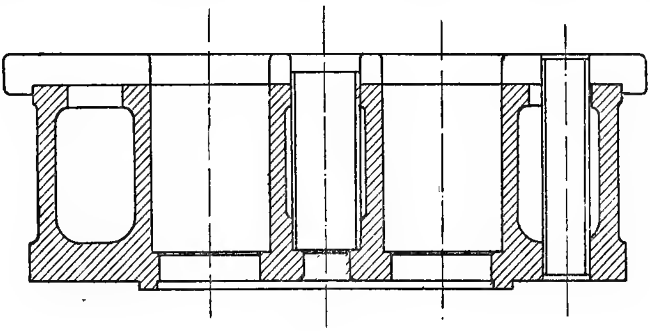


FIG. 108.

appearance facilitates moulding (see Fig. 107). From all considerations, all internal angles should be well radiused.

Water is led to the cover by one of two methods :—

(1) By one or more tubular fittings screwed into the top of cylinder jacket and passing through holes in the under face of the cover (see Fig. 109). With the type of jacket shewn in Fig. 100B it is desirable to fit one such fitting between each pair of cover studs.

(2) By means of an opening in the side of the cover (see Fig. 110).

Whatever means be adopted, it is advantageous to fit internal pipes or baffles to encourage flow towards the fuel valve, as accumulation of deposit at this point is to be avoided at all cost. It is usual to arrange the outlet above the exhaust branch, as stagnation at this point is also undesirable.

Proportions of Cylinder Covers.—The depth of a four stroke cover generally works out to about 0·7 of the cylinder bore, the limiting factors being the size of the exhaust passage and the water space around it. The former should be at least equal in area to the exhaust valve at full lift. The passage starts by being rectangular in shape at the valve end, and gradually becomes circular at the outlet where the diameter is about 0·31 of the cylinder bore. The same applies to the air inlet passage. The thicknesses of metal vary considerably in different designs, and the proportions shown on the sketches represent average practice. The tendency seems to be to

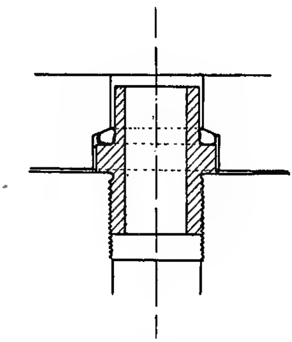


Fig. 109.

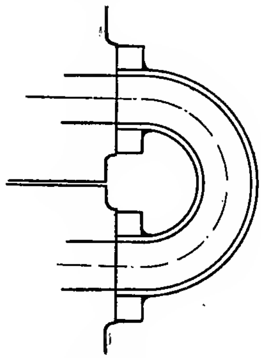


Fig. 110.

rely for safety on a good thickness of metal on the bottom plate.

Strength of Four Stroke Cylinder Covers.—The system of loads acting on the cover comprises the tightening stresses of the studs, the reaction at the spigot joint and the gas pressure on the lower plate. The effect of such a system is to produce tensile stress in the top plate and compression on the bottom. Considering the relative weakness of cast iron in tension and the fact that cracks in the top plate are of very rare occurrence, it would appear that covers proportioned in accordance with average practice have a good margin of safety so far as pressure stresses are concerned. In view of a few isolated failures, or rather as a matter of principle, the strength should be subject to calculation. Owing to the uncertainty as to actual conditions the method of calculation detailed below must be considered comparative rather than absolute.

The assumptions underlying the method are as follows :—

(1) That the severest conditions of stress are due to a cylinder pressure of 1000 lb. per sq. in., due to pre-ignition, careless starting or otherwise, and that this pressure causes the cover to lift to such an extent that the reaction at the joint is eliminated.

(2) That the stress is uniform across a diametrical section in the case of a cover of constant depth and proportional to the distance from the neutral axis of the section in the case of a cover of varying depth. This is not correct, but probably involves approximately equal percentage of error in different cases.

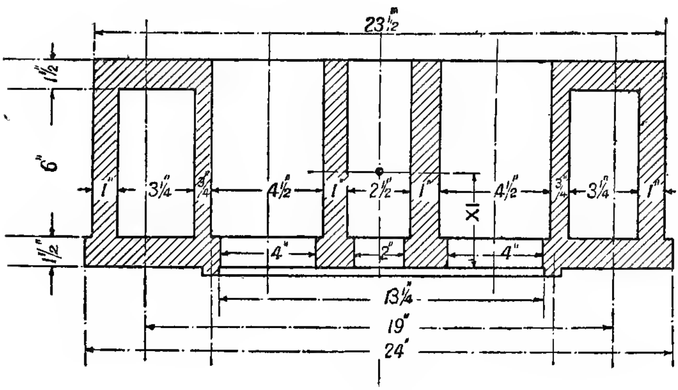


Fig. 111.

Example : Referring to Fig. 111, shewing the weakest section passing through the recesses for the air and exhaust valves, the section modulus is 114 in.^{3}

Considering the forces to the right or left of this section, we have :—

(1) A downward force at the stud circle equivalent to a pressure of 1000 lb. per sq. in. over half the circular area, extending to the joint spigot, viz. : $\cdot784 \times 13\cdot25^2 \times 1000 \div 2 = 69{,}000$ lb. This may be considered to act at the centre of gravity of the pitch semicircle, that is at a distance of $9\cdot5 \times 2 \div \pi = 6\cdot02$ in. from the section under consideration.

(2) An equal and opposite force on the under side of the cover acting at the centre of gravity of the semicircular area extending to the joint spigot, i.e. at a distance of $\frac{6\cdot625 \times 4}{3\pi} = 2\cdot8$ in. from the centre.

The stress is therefore :—

$$\frac{69{,}000\ (6\cdot02 - 2\cdot8)}{114} = 1{,}950 \text{ lb. per sq. in.}$$

Putting the above into the form of a rule :—

$$f = \frac{1000\ R_1^2\ (R_2 - \frac{2}{3}\ R_1)}{z}$$

Where 1000 = Assumed maximum pressure.
f = Stress in lb. per sq. in.
R_2 = Radius of stud pitch circle in in.
R_1 = Inside radius of joint ring in in.
z = Section modulus in in.^{3}

Two Stroke Cylinder Covers.—Where port scavenge is adopted the cover has only to accommodate the following fittings :—

(1) Fuel valve.
(2) Starting valve.
(3) Relief valve (if fitted).
(4) Indicator tube fitting.

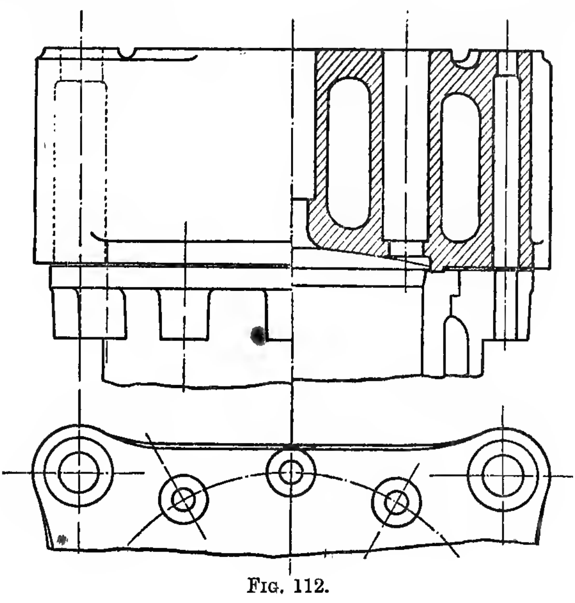


Fig. 112.

As all the above are relatively small the construction of the cover is much simpler than that for a four stroke engine, and failures are rare. In addition, the absence of air and exhaust valves enables the lower plate to be dished upwards (see Fig. 112), an arrangement which gives greater freedom of expansion and a shape of combustion space which is probably favourable to combustion. Fig. 112 shews a cover of this type arranged for four staybolts.

When valves in the cover are employed for scavenging

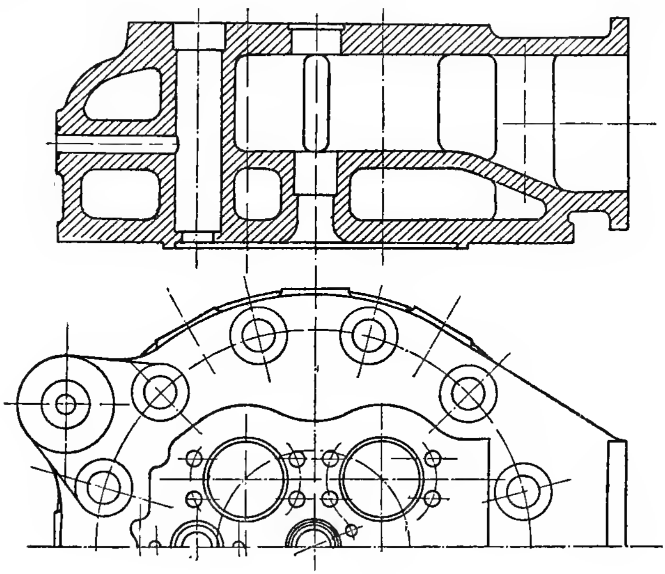


FIG. 113.

purposes the construction depends on the number of valves. If two scavenge valves are used the cover may be of the four cycle type and interchangeable with those of four cycle engines of the same bore. When fitted to a two cycle engine the average temperature of the lower plate will be higher ; but on the other hand the temperature distribution will be more uniform than that of a similar cover fitted to a four stroke engine where one of the valves is used to conduct exhaust gases. The inequality of temperature over the bottom plate of a four stroke cover is strikingly illustrated by the observed fact that in the event of failure the first crack almost invariably runs

between the pockets for the exhaust and fuel valves. A crack between the air valve and the fuel valve sometimes develops at a later stage.

Fig. 113 shews a cover designed to accommodate four scavenge valves. It will be noticed that the interior is divided by a horizontal diaphragm separating the air space and the water-jacket. It appears that this diaphragm and the tubular connections to the bottom plate impose too great restrictions on the expansion of the latter, and fractures have been frequent (with both cast iron and cast steel), so that this type of cover, as hitherto designed, must be considered a failure. The writer understands that a modification of this design (patented by Mr. P. H. Smith), shewn in Fig. 114, has proved satisfactory, and at present no failures have been reported. Apparently the additional depth of the water-jacket, and correspondingly increased freedom of expansion, minimise temperature stresses, and the support afforded by the external shell keeps the bending stresses to a moderate figure.

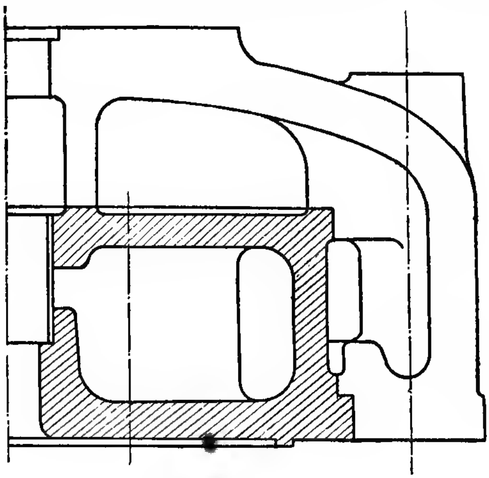


Fig. 114.

Literature.—"Diesel Engine Cylinder Dimensions."—Article in *Engineering*, September 5th, 1913.

Richardson, J.—Paper on Marine Diesel Engines, *loc. cit.*, p. 64.

Hurst, J. E.—"Cast Iron with Special Reference to Engine Cylinders," Manchester Assoc. E., December 9th, 1916.

CHAPTER IX

RUNNING GEAR

Trunk Pistons.—These are so well known in connection with petrol motors, gas engines and the like that a general description is unnecessary, and we may at once proceed to the consideration of their special requirements in Diesel Engine construction. The difficulties involved in combining the piston proper with the crosshead arise chiefly from the heat which reaches the gudgeon pin bearing by conduction and radiation, and the high pressures dealt with in Diesel Engines (as compared with gas engines) necessitate a high specific pressure at this bearing owing to the limited space available. The most serious troubles to be anticipated are piston seizures, which like all other heat troubles are more pronounced in large than in small engines. For these reasons trunk pistons are not generally used for cylinders exceeding about 22 inches in diameter. For marine engines the inaccessibility of the gudgeon pin bearing is usually considered an insuperable objection for all but the very smallest cylinders. It is possible that prejudice has a little to do with this view, and it is interesting to note that Semi-Diesel Marine Engines of fairly large size (500 H.P. for example) apparently give good service with trunk pistons. It is evident that a really efficient system of water cooling would abolish most of the difficulties mentioned. Unfortunately such systems as hitherto fitted to trunk engines have not all been uniformly successful. On the other hand a great measure of success has been achieved in some cases and the ultimate solutions (if not already achieved) are probably near at hand. For crosshead engines water cooled pistons on two or three different systems have already emerged successfully from the experimental stage.

Material.—The use of cast iron for pistons is almost universal on account of its good wearing properties and its cheapness. Owing to the low guide pressure the quality of the metal is

probably of minor importance so far as wear is concerned, as the latter is in any case hardly measurable. On the other hand, so far as that portion of the piston is concerned which is in contact with the working fluid (viz., the piston crown), the quality of the metal is of great importance in determining the liability or otherwise of the crown to crack under the influence of heat. According to Mr. P. H. Smith, the irons which give the best results are those of the coarsest possible grain.

Experience in other directions seems to indicate that a large carbon content and low percentage of phosphorus are favourable.

At the first glance it might appear strange in view of the

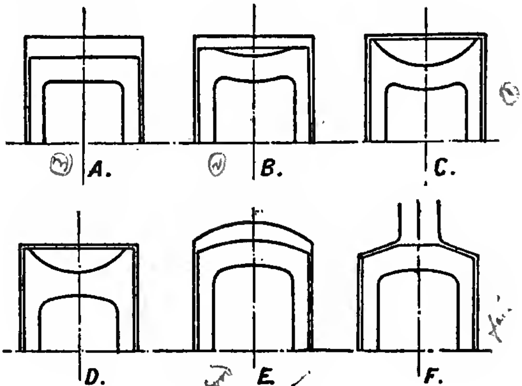


FIG. 115.

fact that heating produces compression at the point of maximum temperature, that the cracks start at the centre of the crown on the side in contact with the gases, but the facts are easily explained by some such hypothesis as the following :—

The local heating causes local compressive stress of high intensity which in course of time causes the particles to rearrange themselves in such a manner that this stress is reduced. On cooling the contraction of the surrounding metal induces tensile stresses in the centre equal in amount to the extent by which the original compressive stress has been reduced. Considerable support is afforded to this theory by the fact that the cracks develop in the course of time into fissures, shewing that the material has contracted circumferentially. Also the plastic deformation of cast iron at a red heat is often observed

in such familiar articles as kitchen ranges and the like, in which no special provision is made for expansion or growth.

Shape of Piston Crown.—Fig. 115 shews some alternative shapes.

Of these types C is the best for combustion, though types A and B are little inferior in this respect. With existing methods of fuel injection, type E is quite inadmissible on the score of combustion unless the cylinder cover is concave downwards, and even then its efficiency is very doubtful. On the other hand, type F, with the shape of combustion space indicated appears to give good results, due doubtless to the manner in which the charge of air is concentrated. A little reflection shews that the curved shape of type B gives rise to greater compressive stress at the centre on the combustion side than type A on account of the bending action which arises when a state of temperature stress comes into existence. Type D is very liable to failure unless a spreading flame plate is used. From this it would appear that of all the types illustrated B and D are the most liable to failure. In practice types B and C are the most usual, and their inherent weakness in the larger sizes (small pistons rarely crack) is guarded against by,

(1) Careful selection of material.
(2) Providing a considerable thickness of metal.
(3) Water cooling.
(4) Providing loose crowns, or central cores.

The provision of a great thickness of metal assists matters in two ways :—

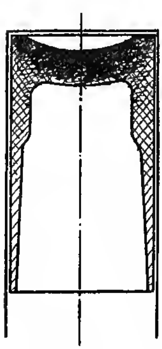


Fig. 116.

(a) By reducing the bending stresses due to both temperature (probably the most vital) and pressure.
(b) By giving additional area for heat flow to the walls of the liner.

Fig. 116 is an attempt to give a diagrammatic representation of the heat flow by increasing the intensity of shading towards the parts having the higher temperatures. The usual form of water cooled

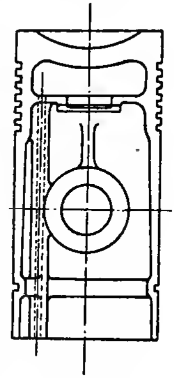


Fig. 117.

trunk pistons is shewn in Fig. 117, and the means adopted to convey the water to and from the water space will be discussed later. In the earlier types the space was intended to be full of water, but in more recent designs the tendency is to rely on a small flow of water, part of which is evaporated and absorbs a relatively large quantity of heat in latent form.

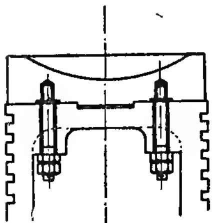


Fig. 118.

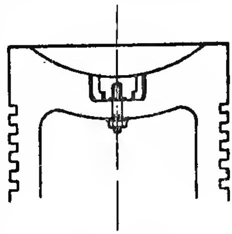


Fig. 119.

Fig. 118 shews a loose piston top secured by four studs, the holes for which have sufficient clearance to allow for the expansion of the former. Fracture of such a loose top due to temperature effects is improbable, and if it occurs the cost of renewal is trivial.

Fig. 119 shews a similar design patented by Mr. P. H. Smith and applied by him to pistons in which cracks had already appeared.

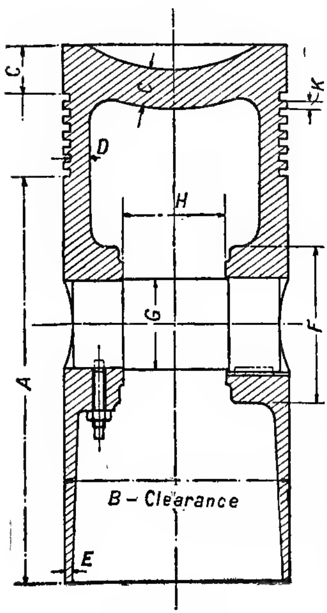


Fig. 120.

Proportions of Trunk Piston Bodies.—The usual proportions found in practice are discussed below with reference to Fig. 120, which is purely diagrammatic.

The thickness (C) of the crown (with uncooled pistons) increases rapidly as the bore is increased, for reasons which have been noticed above, and the following figures are a guide to good practice :—

Bore of cylinder	Thickness C
10″	$1\frac{1}{4}$″
12″	$1\frac{5}{8}$″
14″	$2\frac{1}{4}$″
16″	3″
18″	4″
20″	5″

The distance from the top of the piston to the first ring may conveniently be made equal to C. With

the above values of C this prevents the first ring being placed in too high a position where its proximity to the source of heat would cause it to become stuck with carbonised lubricating oil in a short time. The rings themselves may be of square section with R=0·025 to 0·033 B. The gap between the ends of the ring when free may be about 2·50 times R[1]. The number of rings fitted varies from about five in small to eight in large cylinders, and the space between consecutive rings is not as a rule less than R. The construction of piston rings is rapidly becoming a specialised branch of industry and details will not be given here. A method of designing and manufacturing piston rings is described by Guldner in considerable detail ("Design and Construction of Internal Combustion Engines"). It is important to prevent the joints of the rings working into line during service, and a method of fixing them is shewn in Fig. 121.

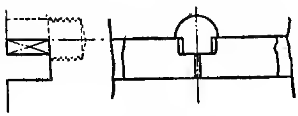


FIG. 121.

Dimension "A" varies greatly in different cases. The maximum value found in practice, viz. A=2B gives a piston of ideal running properties, but is seldom fitted nowadays owing to considerations of first cost. A=1·4B to 1·6B represents good average practice. Still smaller values of A are sometimes used, but are not to be recommended. The provision of a long piston skirt is advantageous from the following points of view :—

(1) Rendering possible a low and therefore comparatively cool location for the gudgeon pin bearing.
(2) It minimises piston knocks.
(3) It facilitates the flow of heat away from the crown by providing a large surface in contact with the cylinder walls.

The upper part of the piston body is turned taper to allow for expansion, and the approximate allowances to be made on the diameter are given in Fig. 122, in which the taper is greatly exaggerated. The diameters of the piston ring grooves are figured by allowing all the grooves to have the same depth below the tapered surface. The clearance behind the grooves

[1] R denotes the thickness of the ring measured radially. The bars K between the rings may be equal to or slightly greater than R.

should be a practicable minimum. The gudgeon pin is usually located either at the centre or slightly above the centre of the parallel part of the body, and allowance is made for possible local distortion due to driving in the gudgeon pin or to ex-

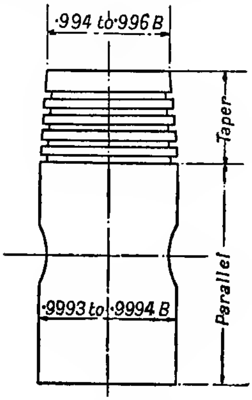


FIG. 122.

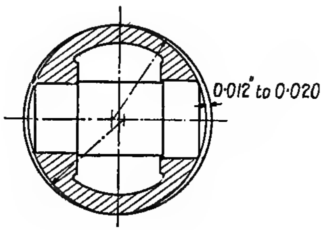


FIG. 123.

pansion of the latter by relieving the piston surface to the extent of about 20-thousandths of an inch, in the manner indicated in Fig. 123.

Approximate figures for other main proportions are given below :—

C (see Table above).	F=2G.
D=0·07B.	G=0·4B.
E=0·033B.	H=0·5B.

Gudgeon Pins.—Alternative forms of gudgeon pins are shewn in Fig. 124, type A being most generally used. The pin itself is of special steel, case-hardened and ground if working in a bronze bearing. If the bearing is white-metalled the case-hardening is unnecessary. The pin is a driving fit in the body and is secured in the manner indicated in the illustrations. Lubrication of the pin may be effected in various ways :—

(1) Forced lubrication by means of a pipe or drilled hole leading from the big end of the rod.

(2) By means of a groove or pocket on the surface of the piston communicating with the surface of the gudgeon pin and fed by means of a fitting similar to that shewn in Fig. 105 (see Fig. 125).

Assuming the proportions given in the previous article, the maximum bearing pressure works out to about :—

$$\frac{0{\cdot}785 \times 500}{0{\cdot}4 \times 0{\cdot}5} = \text{about } 2000 \text{ lb. per sq. in.}$$

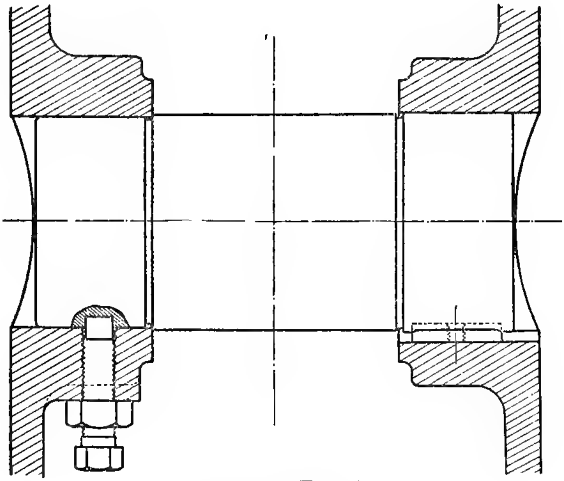


FIG. 124. Type A.

and it is hardly surprising that good bronze appears to be preferable to white metal for this bearing.

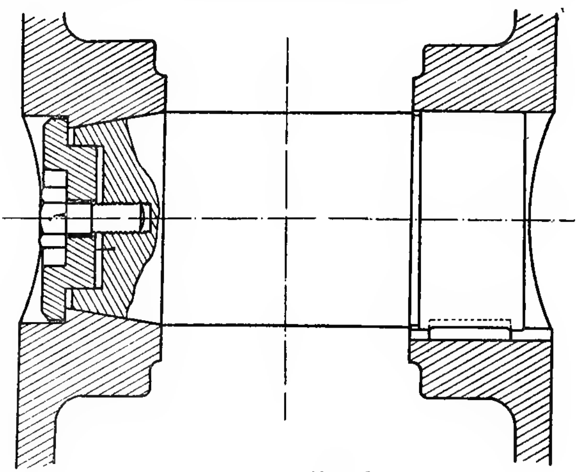


FIG. 124. Type B.

Miscellaneous Points of Detail.—Where forced lubrication is employed it is important to prevent oil from splashing on to the hot piston crown and some arrangement of baffles is very desirable. This may take the form of a plate across the mouth of the piston body or a light guard over the cranks. A piston ring at the lower end of the piston is useful in removing superfluous oil from the liner and in effecting a good distribution of the film.

It is generally necessary to cast two shallow recesses in the

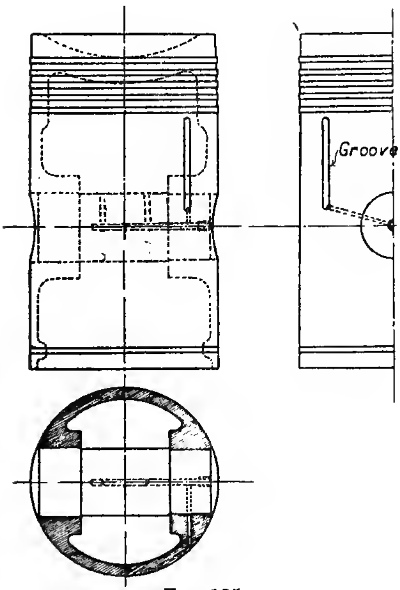


Fig. 125.

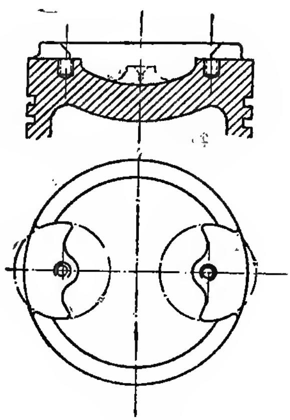


Fig. 126.

piston crown to clear the air and exhaust valve heads at the top dead centre, and the positions of these being remote from the point of greatest temperature are usually chosen for two tapped holes to receive lifting bolts. Holes should not be drilled in the centre of the crown, and if a turning centre is necessary a special boss should be cast for this purpose and turned off afterwards (see Fig. 126).

So far nothing has been said about pressure stresses and the bearing pressure on the piston body considered as a crosshead, as these appear to be irrelevant. Temperature considerations determine the proportions of the piston body, and the gudgeon

bearing is made as large as the limited space allows. The guide pressure between the piston body and the liner works out at a very moderate figure, and measurements of pistons after long periods of service fail to disclose any appreciable wear and justify the conclusion that under normal running conditions the piston body floats on a film of oil. In cases of seizure the cause of abrasion is usually traced to local distor-

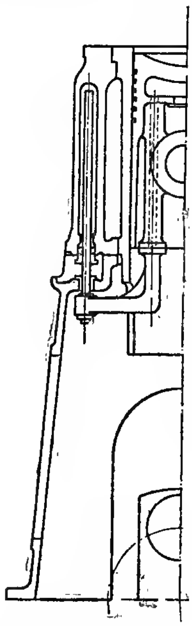


FIG. 127. Type A.

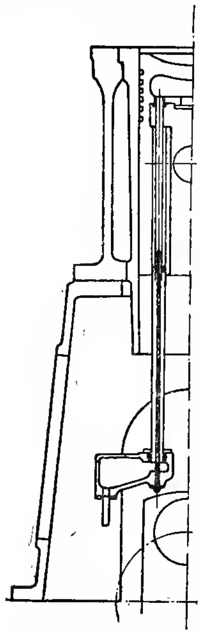


FIG. 127. Type B.

tion, sometimes assisted by the destruction of the oil film by the presence of viscous deposits.

Water Cooling.—Two systems of conveying the water to and from the piston are shewn in Fig. 127. In both systems the aim is to render the success of the scheme independent of the water tightness of the various joints involved. In type B inaccuracies of alignment are allowed for by a ball joint at the foot of the stationary tube.

Pistons for Four Stroke Crosshead Engines.—These are generally made of not much greater length than is necessary to accommodate the rings, eight to ten in number. The pro-

vision of an extra number of rings above what is considered sufficient for a trunk piston may be attributed to :—

(1) The throttling effect lost by discarding the piston skirt.
(2) The lower speed of revolution usually associated with engines of the crosshead type.

Cooling by means of a blast of air has been used (apparently successfully) in cylinders of medium size, but water cooling is now almost universal. Fig. 128 exhibits different forms of piston having one feature in common, viz., ribbed support of the crown. There is reason to believe that these ribs are conducive to cracking in uncooled pistons, and therefore the legitimacy of their use in cooled pistons would appear to depend on the reliability of the cooling system employed.

Assuming that failure of the cooling system is a contingency to be reckoned with, there would appear to be some measure of prudence in proportioning the piston crown to be self-supporting and transmitting the pressure to the piston rod via the piston walls. Two systems of water cooling are shewn in Fig. 129, which requires no description.

Pistons for Two Stroke Crosshead Engines.—The existence of ports in communication with the exhaust pipe at the lower end of a two stroke cylinder necessitates the provision of a skirt or extension of the piston to prevent the uncovering of these ports when the piston is at the top dead centre. The skirt usually takes the form of a light drum secured to the piston by a number of well-locked studs. It is common practice to arrange one or two inwardly expanding rings at the lower end of the cylinder to prevent leakage past the skirt (see Fig. 130).

If the cylinder liner is of sufficient length these exhaust rings may be located in the skirt itself, as in Fig. 131. Such an arrangement involves a higher engine than that of Fig. 130, but facilitates conduction of heat from the piston and incidentally secures a lower mean temperature for the cylinder liner. The construction of the piston proper is generally similar to that of a four cycle engine, but it must be borne in mind that the conditions as to temperature are more severe than in a four stroke engine of the same size working at the same mean indicated pressure, so that the remarks of the preceding article in reference to ribs under the crown apply with still greater force to two stroke engines. It is significant

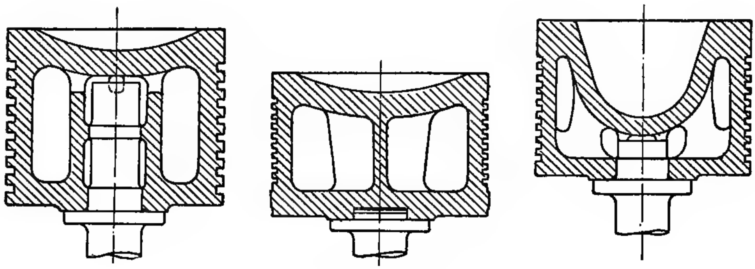


FIG. 128.

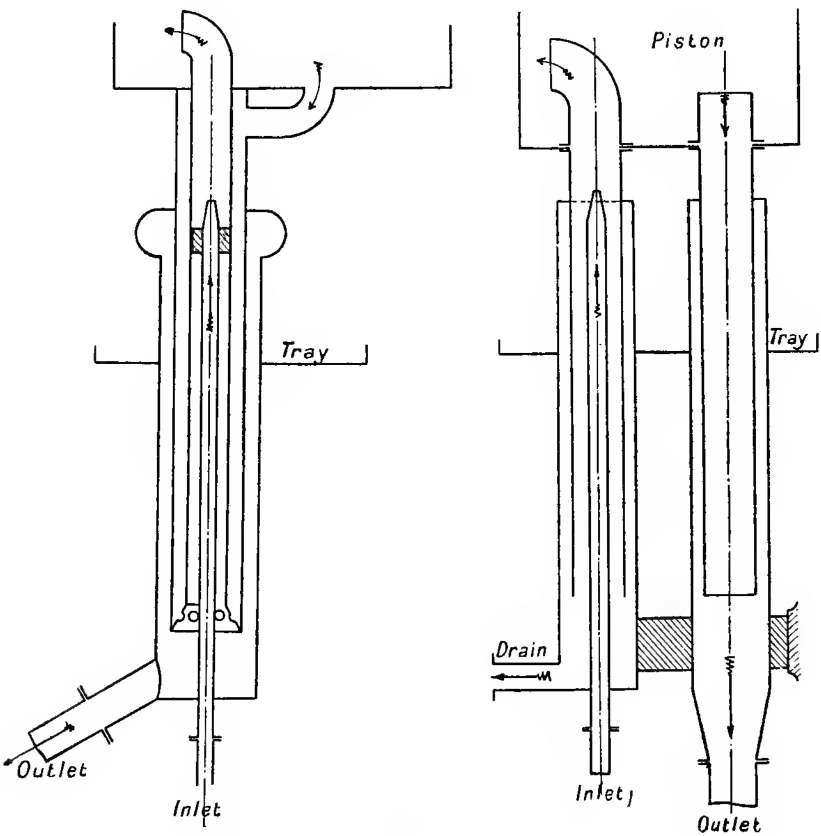


FIG. 129.

that pistons are now being fitted by one of the leading makers of large two stroke engines in which the crown is free from ribs and the skirt is extended to do duty for the piston rod, as shewn more or less diagrammatically in Fig. 132.

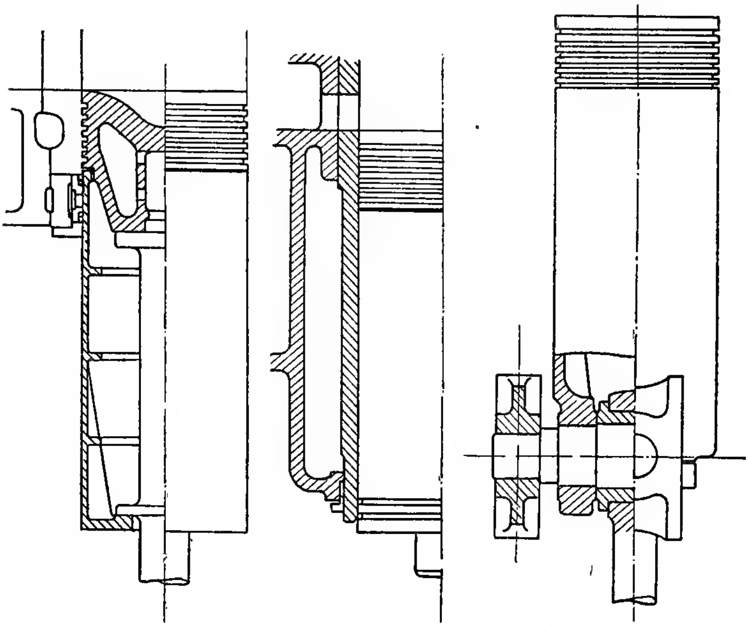


FIG. 130. FIG. 131. FIG. 132.

Piston Rods.—The forces to which a piston rod is subject are :—

(1) The pressure load of the piston, which attains a maximum of about 500 lb. per sq. in. of piston area, with an occasional explosive load of about double this intensity at the top dead centre.

(2) A load due to the inertia of the piston (including the water therein) and the rod itself. This also attains its maximum at the top dead centre, but is opposite in direction to the pressure load. The maximum intensity of the inertia load seldom exceeds 70 lb. per sq. in. of piston area in commercial engines.

(3) Friction of the piston rings on the liner. This effect has its maximum at the top firing centre and acts in the

same direction as the inertia so far as the expansion stroke is concerned. It has been proved that piston ring friction absorbs about 5% of the indicated power, and assuming (as seems probable) that the friction is at every point proportional to the cylinder pressure, it appears that the maximum friction is equivalent to about 10 lb. per sq. in. of piston area.

(4) Fluid friction due to shearing of the lubricating oil film.

The joint effect of these forces amounts then to about 420 lb. compression at firing dead centre and about 70 lb. tensile at the end of the exhaust stroke per sq. in. of piston area.

Unfortunately the fatigue stress of steel between limits of this sort appears not to have been determined yet, but the value is probably in the neighbourhood of 35,000 lb. per sq. in. for 30 ton steel. A table of buckling loads for circular mild steel rods with rounded ends is given below, from which it will be seen that the question of buckling appears to be irrelevant, in view of the fact that the ratio of length to diameter of rod does not usually exceed 15, and is usually less. In this case, and in cases of connecting rods also, the use of a strut formula, such as Euler's, in conjunction with a large factor of safety, would appear to be irrational (see page 184).

L÷D=Length÷Diameter	5	10	15	20	30
Buckling stress, lb./sq. in.	63,000	56,000	45,000	33,000	21,000
L÷D=Length÷Diameter	40	50	80	100	
Buckling stress, lb./sq. in.	13,000	9,000	4,000	2,500	

In current practice the diameter of the piston rod is made from 0·23 to 0·3 of the cylinder bore, corresponding to a maximum compressive stress under normal working conditions of about $\frac{420}{(0 \cdot 23)^2}=8000$ lb./sq. in. to $\frac{420}{(0 \cdot 30)^2}=4700$ lb./sq. in. The higher of these figures corresponds to a factor of safety under fatigue conditions of about 35,000÷8000=4·35, which appears ample. The stress under an explosion of 1000 lb. per sq. in. would be $\frac{1000}{(0 \cdot 23)^2}=19{,}000$ lb./sq. in., and the factor of safety against buckling (assuming $\frac{L}{D}=15)=\frac{45{,}000}{19{,}000}=2 \cdot 35$, which would appear to be sufficient in view of the provision of a relief valve to prevent the pressure from exceeding greatly that for which the rod is designed.

The upper end of the rod usually ends in a circular flange for carrying the piston to which it is secured by a row of studs proportioned to the inertia load of the piston with a very moderate stress allowance, in order to give a good margin for dealing with such emergencies as seized pistons. The lower end is sometimes secured to the crosshead by means of a flange, as in Fig. 133, or in the manner indicated in Fig. 134. In either case it is reasonable to make the connection of the same strength as that between the rod and the piston, assuming that ample provision has been made for the contingencies referred to.

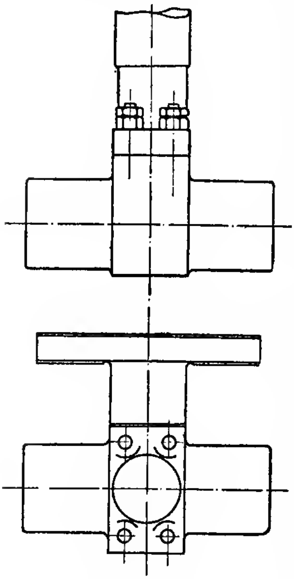


FIG. 133.

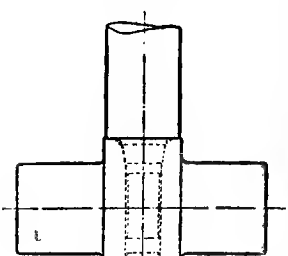


FIG. 134.

Crossheads and Guides. — For marine engines the slipper guide, shewn in Fig. 135, is the favourite. The bearing surface is usually made equal to the piston area,[1] and the maximum bearing pressure with a connecting rod 4·5 cranks long then has a value of about 55 lb. per sq. in. The slipper itself is of cast steel, white-metalled on ahead and astern faces. The studs securing the slipper to the gudgeon block must be adequate to carry the maximum guide pressure when running astern. The area of the gudgeon bearing is based on a bearing pressure of about 1500 lb. per sq. in. The ahead guide face is of cast iron provided with water cooling. The astern bars are frequently of forged steel, secured by fitting bolts. The stress in the latter is usually very moderate, as stiffness is the chief consideration. The proportions given in Fig. 135 are of course approximate only and subject to modification to suit different conditions.

The type of guide block indicated on Fig. 132 is well known

[1] *i.e.* the sectional area of the cylinder.

in connection with paddle-steamers and locomotives, and needs no further description here. For land engines double semicircular guides are sometimes used, particularly when the cylinder and frame are cast in one piece. In general, the cross-

Width of crosshead face, 0·75 B.
Depth ,, ,, 1·10 B.
Thickness ,, plate, 0·10 B.

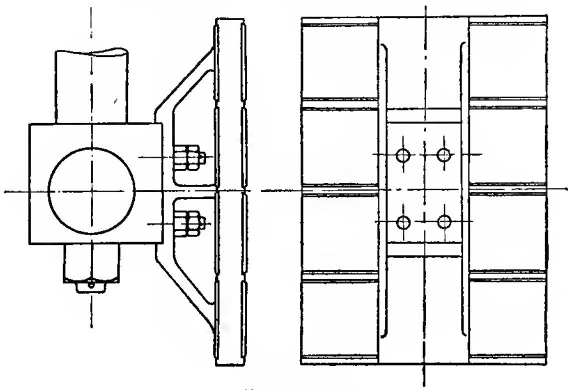


Fig. 135.

heads and guides used in Diesel Engine construction differ but little from those commonly fitted to steam engines and large gas engines, the most important point of difference being the gudgeon pin and its bearing, which require to be liberally dimensioned to withstand the high maximum pressures to which they are subject. It is also desirable to provide means to prevent carbonised oil from the cylinder from reaching the guide surface.

Guide Pressure Diagrams.—A diagram shewing approximately the guide pressure at any crank angle is very simply obtained from the twisting moment curve in the manner described below, with reference to Fig. 136.

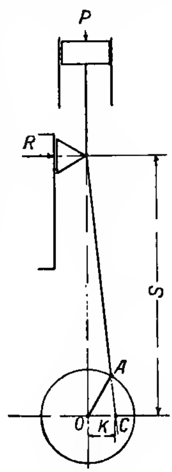


Fig. 136.

P = Piston load in lb.
R = Guide reaction in lb.
T = Twisting moment in in. lb.
S = Height of gudgeon pin centre above the centre of the crank-shaft.

$$\text{Now } R = \frac{P.k}{S}$$

$$\text{But } T = P.k$$

$$\text{Therefore } R = \frac{T}{S}$$

The rule is therefore: Divide the turning moment at any instant by the distance from the gudgeon pin centre to the crank-shaft centre, and the result is the guide reaction at the same instant. It would appear that the guide reaction and the twisting moment should change sign simultaneously. This is not quite the case, for the following reason:—

The twisting moment curve contains an inertia element in which an approximation is obtained by dividing the mass of the connecting rod in a certain proportion between the revolving and reciprocating parts. This approximation, though good so far as vertical forces are concerned, gives very inaccurate values for horizontal forces. Also the centrifugal effect of the revolving parts of the rod influences the guide reaction but not the twisting moment. These discrepancies are of very small importance with the piston speeds at present obtaining. For a full discussion of the influence of the connecting rod inertia forces on the guide reaction, the reader is referred to Dalby's "Balancing of Engines." A table of values of $\frac{S}{l}$ where "l" = the length of the connecting rod, is given below for various crank angles, assuming a rod 4·5 cranks long.

Crank angle	0°	20°	40°	60°	80°	100°	120°	140°	160°	180°
Values of $\frac{S}{l}$	1·22	1·21	1·16	1·09	1·02	0·94	0·87	0·82	0·79	0·78

Connecting Rods.—The material for connecting rods is generally Siemens-Martin steel of the same quality as that used for the crank-shaft. Stampings are sometimes used for small engines, and if large quantities are made at a time this is an economical way of producing a rod of "H" section, if machining all over is not considered essential. Cast steel has been used on the Continent for gas engine connecting rods, but the author has not met with this practice in Diesel Engine construction, either British or continental.

Connecting Rod Bodies.—The section of the body or shaft of the rod is generally circular, or part circular, with flattened

sides. The latter section is slightly lighter for a given strength, but involves an extra machining operation. For extreme lightness an "H" section of rod milled from the solid (like a locomotive rod) would appear to be advantageous. The body usually tapers gently from the big to the small end, and it will be shewn later that this practice is a rational one from considerations of strength. It is interesting to note that the engineers of the North-East Coast, in their recent specification for standard reciprocating marine steam engines, recommend parallel rods, and it seems possible that the extra cost of

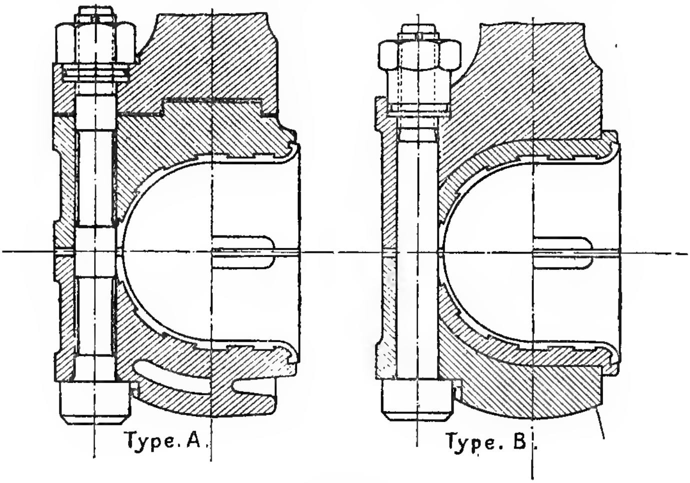


Fig. 137.

material involved is discounted by the reduced cost of forging and machining.

Big Ends.—Fig. 137 shews two forms of big end, type A being the cheapest and that most commonly used. Type B is the strongest, but suffers from the disadvantage of providing no facility for adjusting the compression by means of liners.

Returning to type A, the "brasses" are usually of cast steel lined with white metal. With a stronger section, as shewn in Fig. 138, cast iron may be used instead of cast steel, with satisfactory results, but the practice is uncommon. The bolts are frequently reduced to the core diameter between fitting lengths, as shewn in Fig. 137; but it appears that full diameter bolts are stronger under the conditions to which they are

subject in trunk piston engines. In addition to tensile stresses the big end bolts have to resist shearing forces between the two brasses and also between the crown brass and the palm end of the rod. Partial relief of this duty is afforded by the following means, one or more of which are generally used in good designs :—

(1) Spigoting the two brasses into each other, as in Fig. 139. This is very rarely done.
(2) Spigoting the crown brass into the palm of the rod, as in Fig. 137, Type A.
(3) Providing fitting rings, half in the palm and half in the crown brass at the bolt holes (Fig. 140).

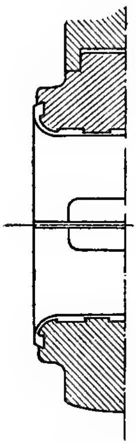


Fig. 138.

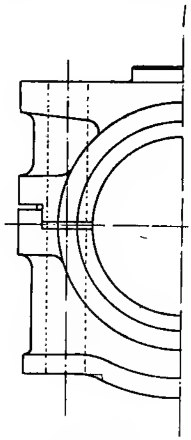


Fig. 139.

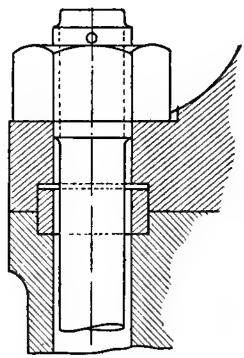


Fig. 140.

Small Ends.—Various types of small end for trunk engine rods are shewn in Fig. 141.

With type A the chief difficulty is to find room for bolts of adequate strength. If a big end of type B, Fig. 137, be fitted, it becomes necessary to make provision at the small end for adjusting the compression, as in Fig. 141, types B and D. Type C combines strength and adjustability of the bearing itself, but makes no provision for altering the compression. Type E contains a solid bush, which must be replaced when worn, and is therefore only suitable for small engines, and adequate section of metal round the bush must be provided to prevent the hole becoming enlarged (see approximate propor-

tions on Fig. 141). The brasses are usually of phosphor bronze.

The forked end of a marine connecting rod is shewn in Fig. 142, on which approximate proportions are noted in terms of the cylinder bore. It differs from the similar member of a marine steam engine chiefly in the following points :—

(1) The cap brass is not provided with a steel keep.

(2) The fork gap is relatively narrower, owing to the piston-rod nut having to deal with inertia only.

(3) The brasses are of cast steel, instead of gun-metal.

Points of Detail.—The lubrication of the big and small ends has been referred to under crank-shafts and pistons. Where forced lubrication is used an oil-hole is generally drilled in the rod to conduct the oil from the big to the small end in the case of high speed engines. External pipes are used for this purpose in large, slow-running engines, but are unsuitable for high speeds owing to the tendency of the joints to work loose.

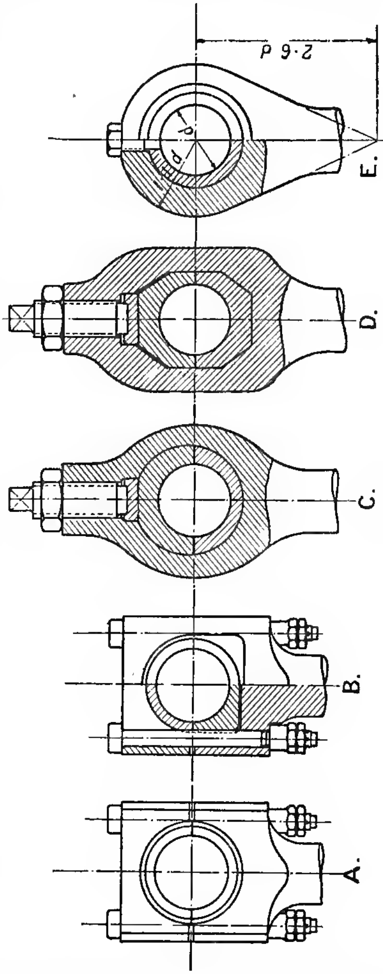


Fig. 141.

When air compressors, oil pumps, or other gear are worked by links from the connecting rod the connection to the latter should be made near the top end, as in Fig. 143, so that the strength of the rod to resist buckling is not impaired.

Strength of Connecting Rods.—The forces acting on the connecting rod are :—

(a) The joint effect of

(1) The pressure load on the piston,

(2) The inertia of the piston and crosshead,

(3) The piston-ring friction,

(4) The lubricated friction of piston and crosshead,

all divided by the cosine of the angle of obliquity of the rod.

(b) The longitudinal component of the inertia of the rod itself.

(c) The transverse component of the inertia of the rod itself.

(d) The friction of the top and bottom end bearings.

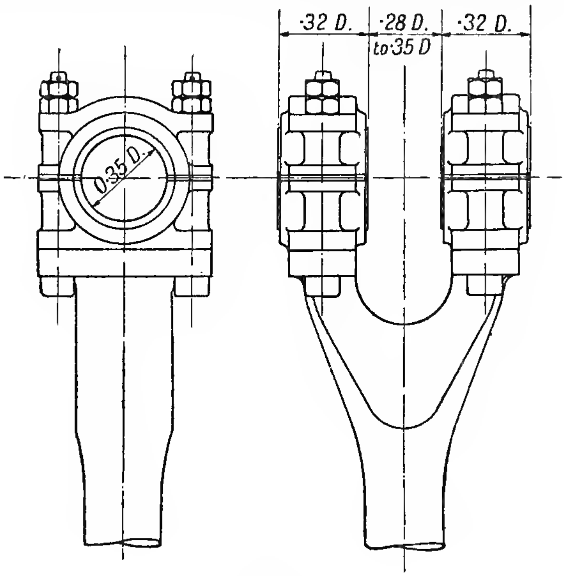


Fig. 142.

When considering the compressive stress of the rod on the expansion stroke one is on the safe side in neglecting item (4). The tensile forces attain their maximum at the top dead centre following the exhaust stroke, and the reciprocating parts being then at their position of minimum speed, item (4) may probably be neglected with safety.

Item (b) is estimated with sufficient accuracy by the usual procedure of dividing the mass of the rod between the recipro-

cating and revolving parts in that ratio in which the centre of gravity of the rod divides its line of centres.

Item (c) gives rise to a bending moment, the maximum value of which is given approximately by the formula :—

$$f = \frac{\left(\frac{n}{100}\right)^2 .R.L^2}{26d} \quad (1)$$

Where f = Bending stress.
n = Revolutions per minute.
R = Crank radius in inches.
L = Length of connecting rod in inches.
d = Diameter of rod in inches (mean).

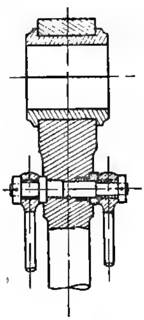


FIG. 143.

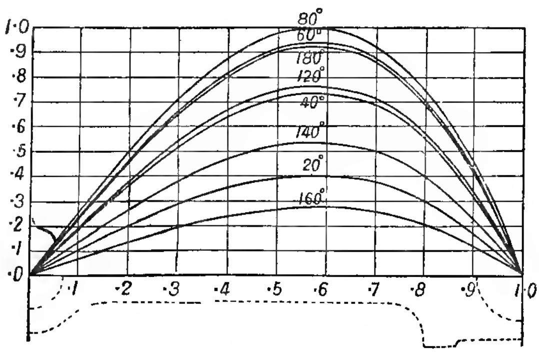


FIG. 144.

The sign of the bending moment is such that the latter always tends to bend the rod outwards to the side on which the crank stands. The variation in the magnitude of the stress over the length of the rod for various positions of the crank relative to the top dead centre is shewn in Fig. 144 for a rod five cranks long. The stress varies as sin $(\theta+\phi)$, where θ = the crank angle relative to top dead centre and ϕ = the angle of obliquity of the rod, and as $Lx - \frac{x^3}{L}$, where L is the length of the rod and x is the distance from the small end to the section under consideration.

The assumptions made use of in equation (1) are that the rod is of uniform section, and, as is usually assumed in books on applied mechanics and machine design, that the influence of the rod ends is small.

Item (d) may be estimated on the assumption that the coefficient of friction attains Morin's value of 0·15 for slightly greasy metal at the top or bottom end or at both ends simultaneously. The effect of journal friction is to divert the line of thrust from the centre line of the rod, and the amount of this deviation is found by the well-known graphical construction shewn in Fig. 145, in which the line of thrust is shewn to be tangential to two very small circles whose radii are

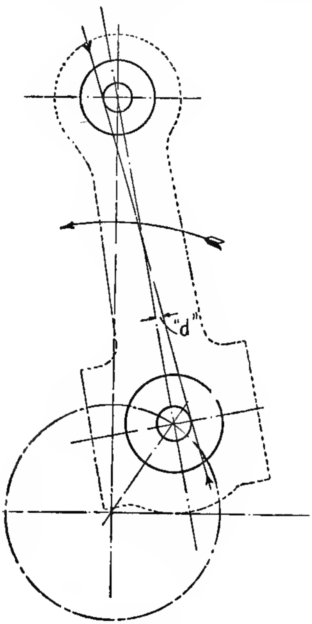


Fig. 145.

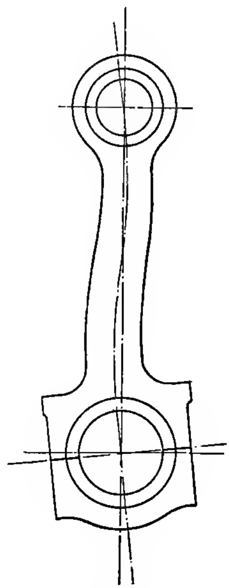


Fig. 146.

equal respectively to the radii of the crank-pin and gudgeon-pin multiplied by the coefficient of friction. The deviation at any point of the rod of the line of thrust from the line of centres will be denoted "d."

The effect of this deviation is to bend the rod into an "S" shape, as shewn much exaggerated in Fig. 146. This form of failure is one consistent with the Eulerian theory of pure buckling, but usually regarded as an improbable solution. The author has actually seen one instance of a Diesel Engine rod failing in this way, and the effect was probably due to the

causes indicated above, arising in acute form. To be on the safe side, it seems advisable to consider two cases :—

(1) Coefficient of friction negligible at the gudgeon-pin and equal to 0·15 at the crank-pin.

(2) Coefficient of friction negligible at the crank-pin and equal to 0·15 at the gudgeon-pin.

The weak sections are clearly those (cf. Fig. 149, KK and SS) where the big and small ends merge into the shaft with a radius, and incidentally those selected by the draftsman when

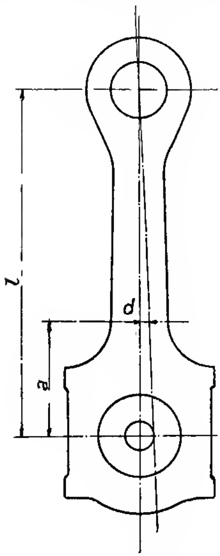


Fig. 147.

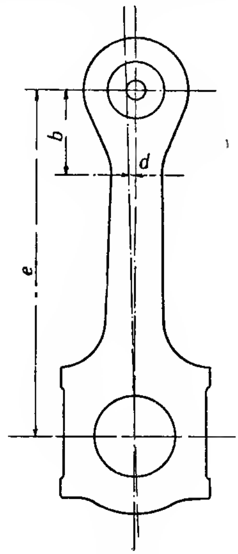


Fig. 148.

giving dimensions for the diameters of the rod. The reasonableness of making the rod tapered will now be apparent, as the bending effect is clearly greater at the large end of the rod on account of the crank-pin being of greater diameter than the gudgeon-pin. In all probability these considerations had nothing to do with the practice, but illustrate a fact that has a very important bearing on the part of machine design, viz., that a construction which appears wrong to the eye of an individual gifted with a sense of form will usually, on investigation be found unsound in principle, and vice versa. This theme is one that might profitably be made the subject-matter

of a less specialised book tan this, but it may be worth mentioning here that the conscious or unconscious recognition of this principle appears to be responsible in part for the ascendancy of continental constructors in certain branches of mechanical design.

Returning to Fig. 146, it is evident that the deflection at any point of the rod should, to be strictly accurate, be added to the deviation of the line of thrust at that point, in order to find the bending moment, and further, this new bending moment involves the construction of a revised deflection curve, and so on. This evidently calls for some form of mathematical treatment, which with certain approximations can readily be applied. It will be found, however, that the deflections involved are small compared with the deviation of the line of thrust, and whatever error may be incurred can be considered to be covered by the factor of safety.

On these assumptions, " d " for the weak sections KK and SS is given by the following :—

Case (1), Fig. 147 . . $d = 0 \cdot 15\, R_c . \frac{l - a}{l}$ ——(2)

Case (2), Fig. 148 . . $d = 0 \cdot 15\, R_g . \frac{l - b}{l}$ ——(3)

And $f = P\left(\frac{1}{A} + \frac{d}{Z}\right)$ ——(4)

Where P = The thrust in the rod in lb.
A = Sectional area of rod in sq. in.
Z = Sectional modulus of rod in in.^3
f = Maximum compression stress in lb. per sq. in.
R_c = Radius of crank-pin.
R_g = ,, ,, gudgeon-pin.

It will be noticed that the strut formulæ of Euler, Gordon and Rankine and others have not been utilised above. It appears to the writer that these formulæ are irrelevant to the case of Diesel Engine connecting rods, for the following reasons :—

(1) Euler's formula is based on the calculation of the load required to produce elastic instability, and with short rods the stress commonly works out at a higher value than the ultimate strength.

(2) The Gordon and Rankine formulæ are based on experimental values of the buckling stress under static

conditions, and give no indication of the strength under repetitions of stresses, which are generally only a fraction of the buckling load.

It seems more rational, therefore, to calculate the maximum direct stresses as closely as possible and to apply to the approximately known fatigue stress of steel a factor of safety of 2·5 to 3, which is known to be satisfactory in other cases.

In view of occasional abnormal pressures of about 1000 lb. per sq. in., it is interesting to see what factor of safety a given rod has for meeting such contingencies, and the table of buckling stresses given on page 173 may be used for this purpose.

Example of Stress Calculation for Connecting Rod.—

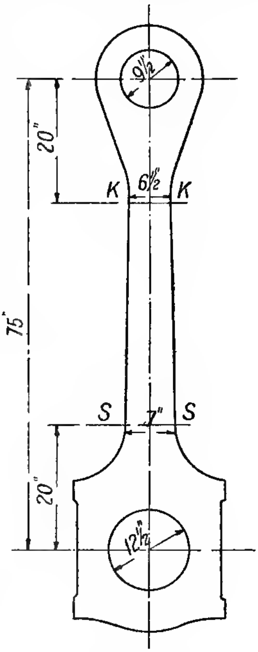


Fig. 149.

Four Stroke Cycle.

Bore of cylinder . .	24 in.
Stroke.	30 in.
Revolutions per minute	200
Weight of piston (trunk)	2200 lb.
,, connecting rod complete . . .	2500 lb.

Main dimensions of rod, as in Fig. 149, under :—

(1) Calculation of stress due to thrust 30° after firing centre.

Piston pressure load $=0\cdot785\times24^2\times500=226{,}000$ lb.

Inertia load 30° after dead centre

$$=\left(\frac{2\pi\times200}{60}\right)^2\times15\times\frac{2200+0\cdot35\times2500}{386}\times(\cos 30^\circ+\tfrac{1}{5}\cos 60^\circ)$$

$$=51{,}000 \text{ lb.}$$

Resultant vertical force $=226{,}000-51{,}000=175{,}000$ lb.

At 30° after dead centre the obliquity of the rod is 6°.

$\therefore$ Connecting rod thrust $=175{,}000\div\cos 6^\circ=176{,}000$ lb. $=$P.

At section KK :—

Area $=33\cdot2$ in.$^2=$A

Section modulus $=27\cdot0$ in.$^3=$Z

Deviation of line of thrust $=\dfrac{0\cdot15\times4\cdot75\times55}{75}=0\cdot52=d$

and the stress $f=176{,}000\left(\dfrac{1}{33\cdot2}+\dfrac{0\cdot52}{27\cdot0}\right)=8970$ lb. /sq. in.

At section SS :—

$$\text{Area}=38\cdot5 \text{ in.}^2=A$$
$$\text{Section modulus}=33\cdot7 \text{ in.}^3=Z$$

Deviation of line of thrust $=\dfrac{0\cdot15\times6\cdot25\times55}{75}=0\cdot69$ in.

and the stress $f=176{,}000\left(\dfrac{1}{38\cdot5}+\dfrac{0\cdot69}{33\cdot7}\right)=176{,}000\times0\cdot0464$

$$=8160 \text{ lb. /in.}^2$$

(2) Calculation of stress due to inertia bending at 30° after dead centre.

Maximum inertia stress in rod, from equation (1),

$$=\frac{2^2\times15\times75^2}{26\times6\cdot75}=1920 \text{ lb. /in.}^2$$

From Fig. 145 the fraction of this maximum applying to postion KK at 30° after dead centre is 0·37,

$\therefore$ Inertia bending stress at section KK $=0\cdot37\times1920$

$$=710 \text{ lb./in.}^2$$

The fraction applying to section SS at 30° after dead centre is 0·49.

$\therefore$ Inertia bending stress at section SS $=0\cdot49\times1920$

$$=940 \text{ lb. /in.}^2$$

(3) Resultant stress at KK $=8970-710=8260$ lb. /in.^{2}

Since the bending actions due to inertia and eccentricity of thrust are of opposite sign,

Resultant stress at SS $=8160+940=9100$ lb. /in.^{2}

since the two bending actions are of the same sign.

(4) Tensile stress at SS at beginning of suction stroke.

$$\text{Inertia force}=\left(\frac{2\pi\times200}{60}\right)^2\times15\times\frac{2000+0\cdot35\times2500}{386}\times1\cdot2$$
$$=63{,}200 \text{ lb.}$$

$$\text{Stress at SS}=63{,}200\left(\frac{1}{38\cdot5}+\frac{0\cdot69}{33\cdot8}\right)=2930 \text{ lb. /in.}^2$$

The total range of stress is therefore

$$9100+2930=12{,}030 \text{ lb. /in.}^2$$

The range of stress required to produce fracture of mild steel by fatigue appears to be about 35,000 lb. /sq. in., so the factor of safety is about 3.

Calculation on the above lines might with advantage be made for several different positions of the crank.

It is evident that the results of the calculation depend very largely on the assumed conditions of journal friction, but it should be borne in mind that almost any possible combination of unfavourable conditions is a probable contingency in the combined lives of a number of similar engines.

Proportions Found in Practice.—In the preceding example the mean diameter of the rod is approximately 0·28 of the diameter of the cylinder, a very favourite ratio in practice. In different designs this ratio varies from 0·26 to 0·30, and it is a rather curious fact that the two extreme figures are those which appear to be used by two of the leading makers in their respective practices.

The maximum and minimum diameters are usually about 5% more and less than the mean.

Connecting Rod Bolts.—In four stroke engines these are usually proportioned to the maximum inertia load with a nominal stress of 6000 to 8000 lb./in.^{2} based on the inertia and centrifugal loads divided by the area of two bolts at the bottom of the threads. With trunk piston engines failure when it occurs is generally due to piston seizure, to which it would be difficult to apply definite rules of calculation. Danger of seizure is largely eliminated by the use of a cross-head. The strength of connecting rod bolts for four stroke Diesel Engines forms the subject-matter of a paper by Mr. P. H. Smith, read by him before the Diesel Users' Association and containing the results of several years' experience. For the big end it appears that the bolts seldom fail if made of a diameter 12 to 13% of the cylinder bore. For the small end, if bolts are used at all, the only safe rule is to make the bolts as large as the space available will allow. Mr. Smith also points out that the bolts, for both big and small ends, are not equally stressed, as may easily be seen by reference to Fig. 150.

Owing to the deviation of the line of pull from the centre line of the rod, that bolt (No. 1 in the Fig.) which first passes the top dead centre at the beginning of the suction stroke is nearer the line of pull than the other bolt, and consequently more highly stressed.

If P = Resultant pull in lb.
S = Centres of bolts in in.
d = 0·15 radius of crank-pin.

Then

$$\text{Pull in bolt No. 1} = \frac{P \times \left(\frac{S}{2} + d\right)}{S}$$

$$\text{Pull in bolt No. 2} = \frac{P \times \left(\frac{S}{2} - d\right)}{S}$$

With crosshead engines the small end bolts have to carry the inertia load due to piston, piston rod and crosshead, and also any frictional forces acting on the piston and crosshead. The

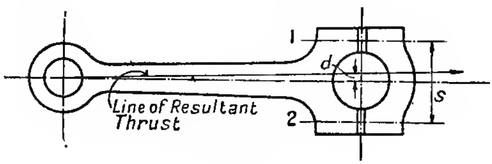


Fig. 150.

latter being more or less indeterminate, it is customary to allow a nominal stress on these bolts about 30% less than that allowed for the big end bolts. The bolts or studs connecting the crosshead to the piston rod and the latter to the piston are given a large margin of strength for the same reason. Connecting rods for two stroke engines are not as a rule distinguishable from those for four stroke engines, as the possibility of compression being lost has always to be kept in view.

Indicating Gear.—The only satisfactory gear for obtaining accurate cards consists of a link motion directly connected to the piston. The usual arrangement is shewn diagrammatically in Fig. 151 for both trunk and crosshead engines. The conditions for giving an accurate reproduction of the motion of the piston are :—

(1) The line of the cord to be at right angles to the mean position of the short arm of the lever.
(2) The long arm of the lever to make equal angles of swing above and below the horizontal.
(3) The versine of the arc of swing of the drag link to be negligible in comparison with the stroke of the engine.

The mechanical details of indicator gears are hardly of sufficient interest to require description here, but it may be well to mention that indicating is of far more importance in the successful running of a Diesel Engine than in that of a steam or even a gas engine, and consequently all the more care should be given to the design of the gear by which the indicating is accomplished. Makeshift or temporary gear should not be tolerated, but the same attention paid to lubrication and bushing of joints, etc., as on other parts of the engine.

Literature.—"Piston Cooling for Diesel Engines." Article in *The Motor Ship and Motor Boat*, July 18th, 1918, *et seq.*

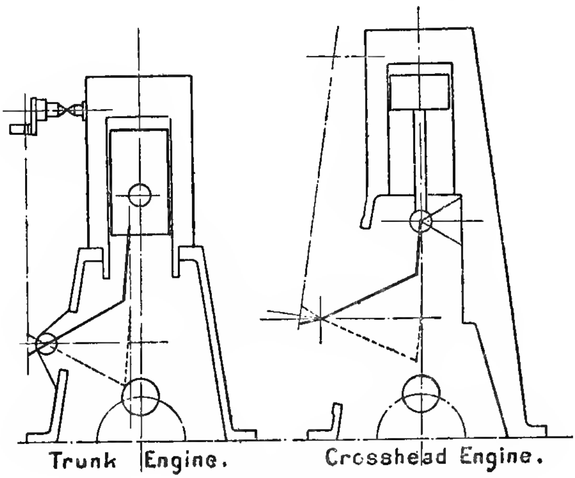


FIG. 151.

CHAPTER X

FUEL OIL SYSTEM

For purposes of description the complete fuel oil system is conveniently divided into two parts, the first consisting of those elements, such as tanks, etc., which are external to the engine, and the second of those organs of the engine itself which are directly concerned with the delivery of the fuel to the working cylinder.

External Fuel Oil System.—Fig. 152 represents diagrammatically a fuel system for a small Diesel power station and

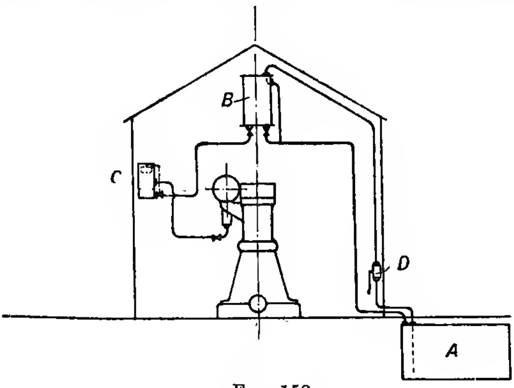


Fig. 152.

consists essentially of a main storage tank A, a ready-use tank B, a filter C, and a pump D, for raising the oil from the storage tank.

The storage tank is preferably arranged underground, as close as possible to the railway siding, so that oil can be run from the railway tank waggon to the storage tank by gravity, through a hose pipe. Some form of level indicator or a plugged hole for a sounding rod should be provided. The capacity of the tank will depend on the size of the station and the local conditions of supply.

The pump D, by means of which the oil is pumped to the ready-use tank, may be of the semi-rotary type, capable of being worked by one man in the case of small stations; but where the daily demand is greater, a motor-driven rotary or reciprocating pump is generally fitted.

The ready-use tank may have a capacity of say half a day's run, so that the routine of replenishing it will occur twice daily. Some form of float indicator should be fitted, so that the level of oil may be conveniently ascertained from the engine room floor. Other necessary fittings are an overflow pipe leading to the storage, or a special drain tank, and a drain valve communicating with the overflow pipe. The tank must be closed at the top to exclude dirt. The valves in connection with the fuel system are preferably of the gate or sluice type, as cocks are liable to leakage and globe valves tend to choke by accumulation of sediment. The tank only requires to be located a few feet above the level of the filter as the discharge is very small.

The filter usually consists of a cylindrical tank of about 40 gallons capacity, located about two feet above the level of the cylinder cover and provided with a filter diaphragm at about a third of the height of the tank from the bottom. The diaphragm consists of a sheet of felt sandwiched between two sheets of wire gauze and reinforced by an angle iron ring.

The fuel enters the filter at the bottom, passes through the diaphragm by virtue of its static head, and is drawn off by the engine fuel pump at a point a few inches above the diaphragm. The filter vessel is prevented from being overflooded by a ball float mechanism which closes the inlet cock when the oil reaches a predetermined level. The plug of this cock is kept fairly tight by means of a spring acting on the plug, but slight leakage is almost inevitable, so it is desirable to mount the filter on an oil-tight tray provided with a drain. It is very usual to provide a small reservoir of the same capacity as the filter arranged alongside the latter for the reception of paraffin, by means of which the piping leading to the engine, and also the fuel pumps and fuel valves, etc., may be cleansed from time to time by running the engine for a few minutes on this fuel before stopping the engine.

Marine installations follow on similar lines with a few complications. The double bottom is used as a storage tank, and the fuel is raised to the ready-use tank by motor-driven pumps, when electric power from auxiliary engines is continuously

available or by means of pumps driven off the main engine in cases where the main engine drives its own auxiliaries. In either case it is usual to install duplicate pumps to guard against breakdown. The motion of ships being unfavourable to the successful operation of float devices, the level of the oil in the ready-use tank has to be inferred from gauge glasses, test cocks and the like. For similar reasons, the filters must be totally enclosed and provided in duplicate with change-over cocks, so that they may be overhauled at any time. In addition, special requirements of the Board of Trade and Lloyd's have usually to be complied with.

Fuel System on the Engine itself.—The commoner arrangements fall into one of two broad classes :—

(1) Those in which each cylinder has a separate fuel pump or separate plunger and set of valves to itself. In this case the oil is delivered direct from the pump to the injection valve by the most direct route possible.

(2) Those in which one fuel pump plunger supplies the oil for a plurality of cylinders, usually a maximum of four. In this case the pump delivers to a fuel main provided with a branch and distributing valve separate to each cylinder, whereby the amount of oil delivered to each cylinder may be equalised while the engine is running.

Engines of six or eight cylinders are divided into two groups of three or four respectively, so far as the oil system is concerned. For marine engines the first system is at present the favourite, and has the advantage that the failure of one pump does not affect the working of the remaining cylinders. With the second system a stand-by pump is provided, ready to take over duty at a moment's notice.

For land engines the two systems appear to be on an equality as nearly as can be ascertained by the reputations of representatives of both classes, but if anything system (1) appears to be slightly the more popular of the two.

Figs. 153 and 154 illustrate the two systems diagrammatically. It will be noticed that in Fig. 153 the governor operates on all the pumps by means of a shaft extending nearly the whole length of the engine, and as the quality of the governing is dependent on the freedom from friction of the governing mechanism it is desirable to mount this shaft on ball bearings. The expense of providing separate pump bodies

and drives is sometimes reduced by grouping the pumps in the neighbourhood of the governor, even to the extent of driving all the plungers by a common eccentric.

With the arrangement of Fig. 154 there is only one pump to regulate, and this renders possible the use of a type of governor

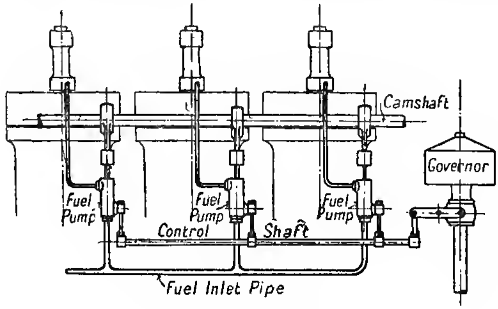


FIG. 153.

which is probably unrivalled for sensibility and which will be described later. The distributors indicated in Fig. 154 are a special feature of this system and are illustrated to a larger scale in Fig. 155. A particularly neat arrangement of piping is obtained by combining the fuel distributor and blast air T-piece in one fitting.

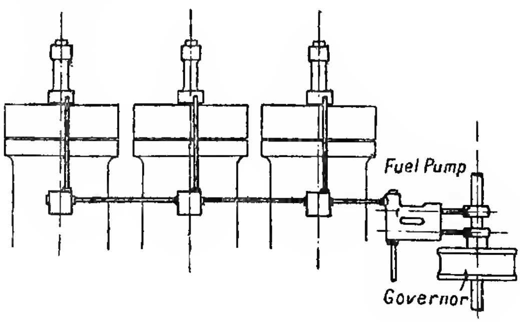


FIG. 154.

The inclusion of a non-return valve prevents in a great measure the oil being forced back through the pump by the blast air pressure in the interval elapsing between the turning on of the blast air and the attainment by the engine of full working speed. A non-return valve is sometimes fitted to the

fuel valve itself for the same reason. Vent cocks are provided on the distributors, and sometimes on the fuel valves, to enable the pipes to be primed before starting the engine.

The priming may be effected in various ways :—

(1) By gravity, means being provided for holding the fuel pump valves off their seats during the process.

(2) By means of an auxiliary hand-operated and spring-returned plunger on the fuel pump.

(3) By means of the fuel pump plunger itself, where provision has been made for disconnecting the latter from its operating eccentric in order to enable it to be operated by a hand lever provided for the purpose.

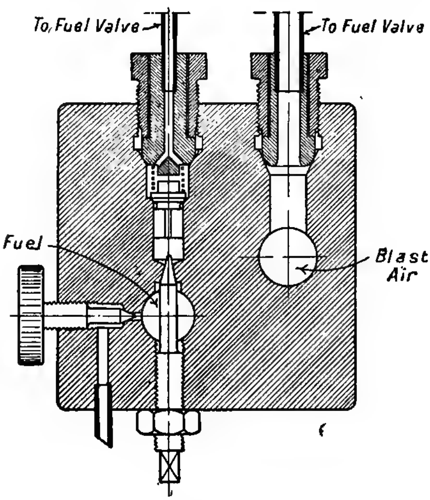


Fig. 155.

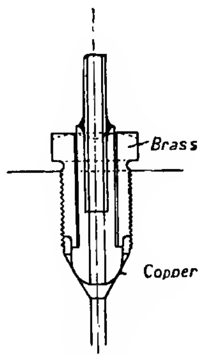


Fig. 156.

The piping in connection with the high pressure fuel system deserves special attention, on account of the high pressures used, and the type of union shewn in Fig. 156 is probably the most satisfactory that has yet been devised both for oil and high pressure air.

Fuel Pumps.—A simple fuel pump for a large marine engine is shewn in Fig. 157, and is representative of a large class of pumps for both marine and stationary purposes.

The operation of the pump is almost obvious from the figure, but may be described briefly as follows :—

The eccentric A works the plunger B, which is guided at C. E and F are the delivery and suction valves respectively, and the latter communicates with the suction chamber D, to which the fuel is led by means of a pipe not shewn in the figure. M is an auxiliary plunger operated from the crosshead C by links I, H, K, etc., and whose function is to keep the suction valve off its seat for a fraction of the delivery stroke, depending upon how much oil is required per stroke. The duration of this inoperative portion of the stroke is altered as required by raising or depressing the point K by means of an eccentric keyed to the shaft L, according as less or more oil is required. In the case of a governed engine the shaft L is controlled by the governor. On marine engines the shaft L is operated by hand gear, consisting of levers, rods, etc. Neglecting the obliquity of the eccentric rod the main plunger describes simple harmonic motion of amplitude equal to half the stroke of the eccentric, and it will be clear from the drawing that the auxiliary plunger M will describe a similar motion exactly in phase with the first but of amplitude equal to stroke of main plunger $\times \frac{KH}{KI}$

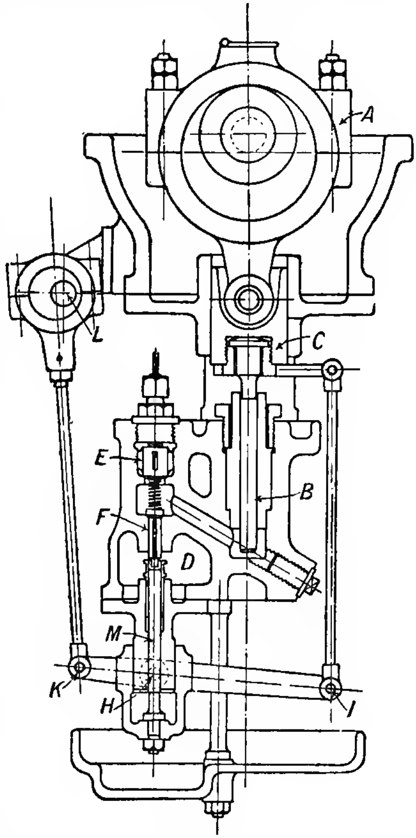


FIG. 157.

When the main plunger is at the bottom of its stroke the auxiliary plunger is also at the lowest point of its travel, and the clearance between the top of the auxiliary plunger and the suction valve multiplied by the ratio $\frac{KI}{KH}$ is equal to the effec-

tive stroke of the pump, that is that portion of the stroke during which the suction valve is on its seat, as of course it must be (apart from viscosity effects) for delivery to take place. The quantity of oil delivered per stroke therefore depends on a certain clearance between the auxiliary plunger and the suction valve, which clearance is readily adjusted by shortening or lengthening the rod LK when assembling or adjusting the engine and in the ordinary course of running by the eccentric at L.

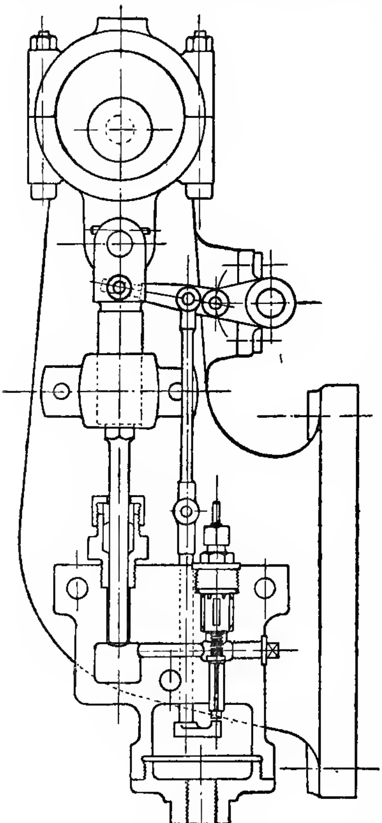


Fig. 158.

Constructional Details.—The pump body, plunger sleeve and guide are of cast iron. The main and auxiliary plungers, the crosshead pin and joints in the linkwork are of case-hardened steel. The valves may be either of steel or cast iron. If the latter, then the suction valve should be fitted with a hardened steel thimble where it makes contact with the auxiliary plunger. The main eccentric and strap may be of cast iron, the lower half of strap being white-metalled in some cases. It will be noticed that no packing is provided for the main plunger but reliance has been placed on the fit of the plunger. With good workmanship the leakage should not be excessive.* A cast tray is provided to catch drips during working and the overflow at priming. A light sleeve encircling the auxiliary plunger is arranged for operation by external gear so that the suction valve may be lifted by an emergency governor in cases of excessive speed, and also by hand in case it is desired to cut any individual cylinder out of operation.

Variations of this system, embodying the same principle

* Except with tar oil, for which this arrangement is unsuitable.

are shewn in Figs. 158 and 159. The front view of the latter shews three pumps grouped together, but each worked by its own eccentric. Fig. 160 shews four plungers being operated by eccentrics in common. It is evident that with this arrangement the oil delivered to the pulverisers of the various cylinders will have different allowances of time in which to settle before injection into the cylinder. This appears to have no effect on the efficiency, but it is usual to space the eccentrics so that oil is not in process of delivery whilst a fuel valve is open. The pumps so far illustrated have been driven off the cam-shaft. That shewn in Fig. 161 is arranged with horizontal plungers

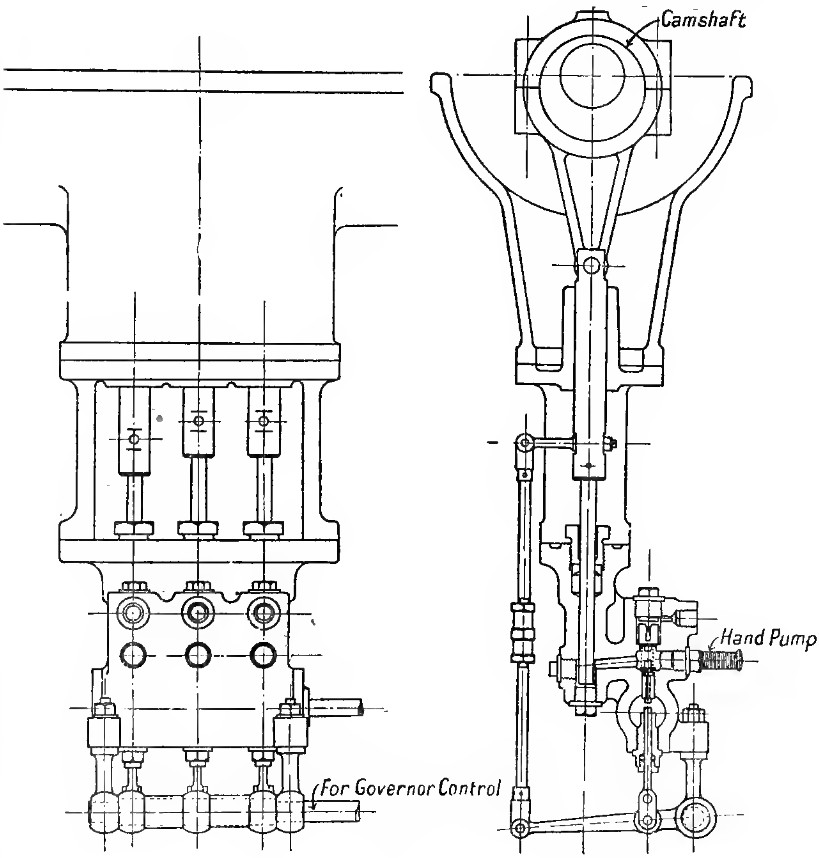


Fig. 159.

for driving off a vertical shaft. The auxiliary plunger is driven by a separate eccentric which on account of the intermediate lever L requires to be at 180 degrees or thereabouts to the main eccentric. The suction valve control may in this case be effected in one of two ways.

(1) By an eccentric movement of the lever L.
(2) By advancing or retarding the auxiliary eccentric.

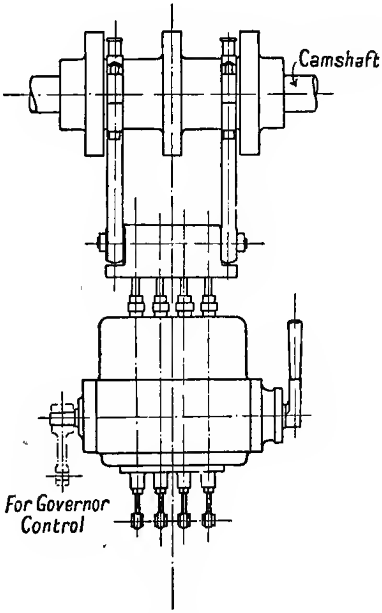


Fig. 160.

The latter leads to a very neat and efficient arrangement of governor and fuel pump, to which reference has already been made. It will be immediately obvious that with a given maximum clearance between the suction valve and the auxiliary plunger, an angular movement of the auxiliary eccentric will have the effect of advancing or retarding the instant at which the suction valve comes on its seat, and consequently increasing or decreasing the amount of oil delivered per stroke. This angular movement is effected very simply by a type of governor which has been well known for a long time, in steam practice, and which is illustrated in Fig. 170.

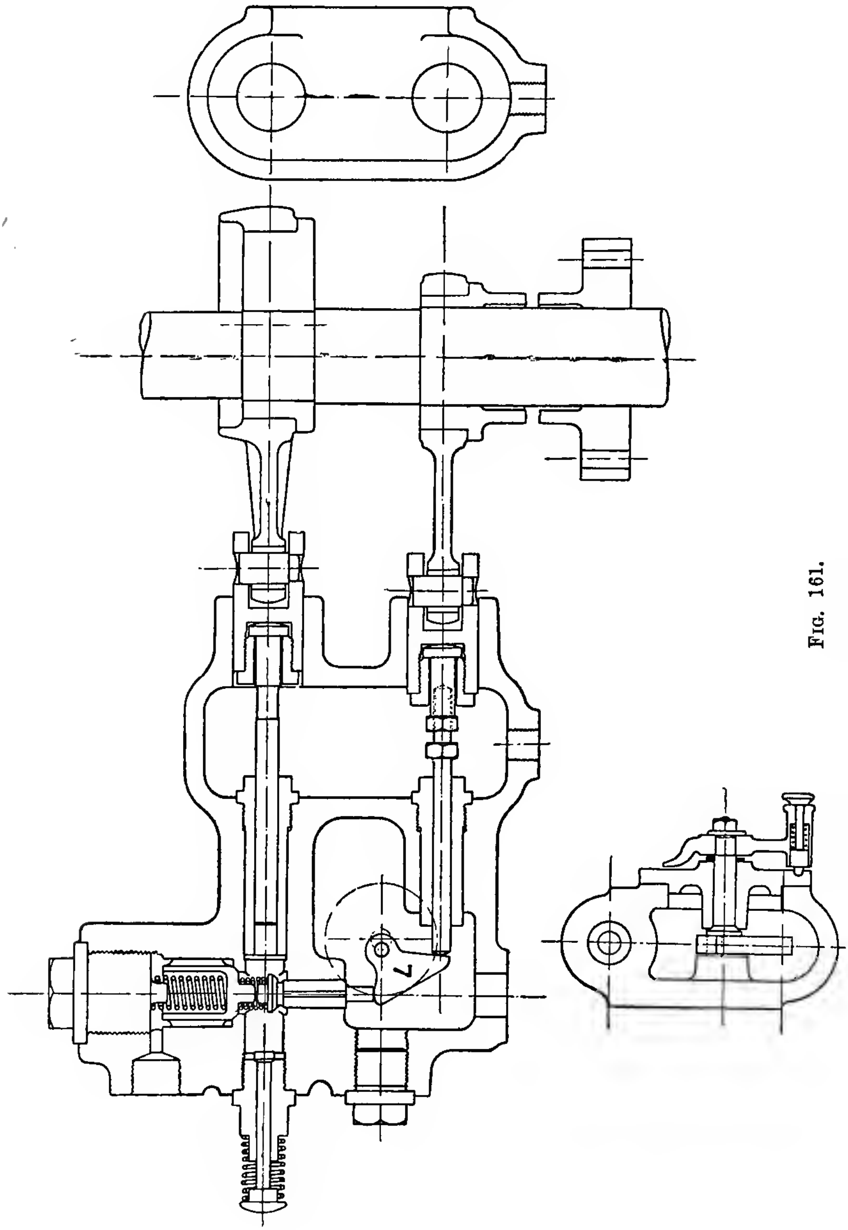


FIG. 161.

Returning to Fig. 161, the use of this type of fuel pump is almost entirely confined to land engines. The provision of an eccentric mounting for lever L enables the pump to be set in three different positions, apart from its normal running position, viz. :—

(1) "Starting." In this position the lever is moved so that the maximum clearance under the suction valve is increased about 50%, so that the delivery of oil per stroke is increased correspondingly.
(2) "Stop." In this position the suction valve is held continuously off its seat and no oil is delivered.
(3) "Priming." In this position both suction and delivery valves are held off their seats and the oil has a clear passage through the pump.

Figs. 163 and 164 will make this matter clear without further explanation.

Details of Fuel Pumps.—The bodies are usually of cast iron, but solid slabs of steel are sometimes used. In designing the body three considerations should be kept in view :—

(1) The shape to be favourable to sound casting.
(2) As few machining operations as possible to be necessary apart from those which can be done on a drilling machine.
(3) The pump chamber and passages to be free from air-locks.

Owing to the costly precautions necessary to ensure the plunger and guide being concentric and in line it is convenient to allow some side play at the point where they join, as in Fig. 165. Some different forms of plunger packing are shewn in Fig. 166 and a selection of suction and delivery valves in Fig. 167. Fig. 168 shews a hand-operated plunger for priming purposes.

Calculations for Fuel Pumps.—The process of computing the capacity of a fuel pump for a proposed engine is most easily illustrated by an example, as follows :—

B.H.P. of one cylinder (four stroke), 250.
One plunger to each cylinder.
Estimated fuel consumption, 0·4 lb. per B.H.P. hour.
Revolutions per minute, 120.

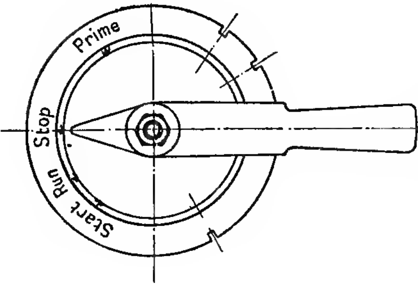


FIG. 163.

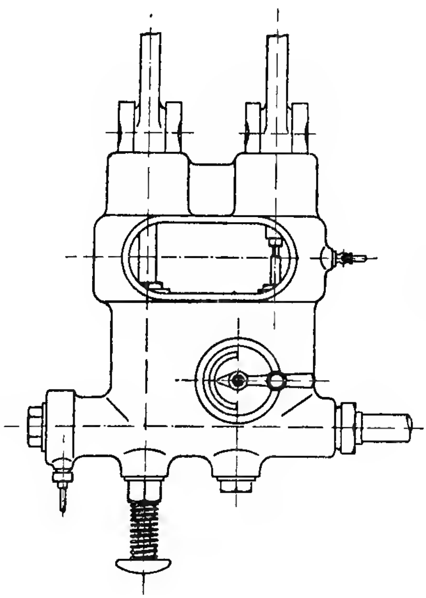


FIG. 162.

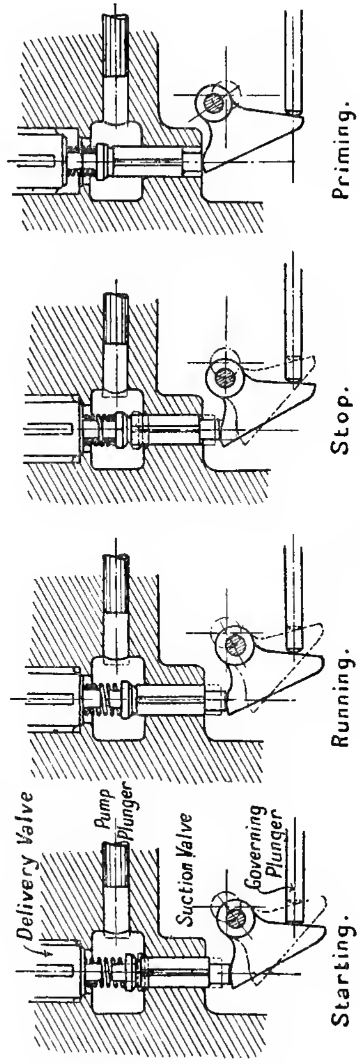


FIG. 164.

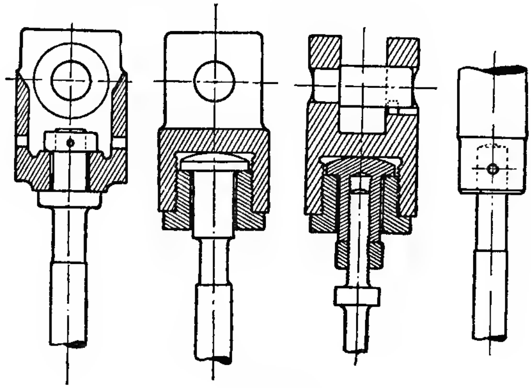


FIG. 165.

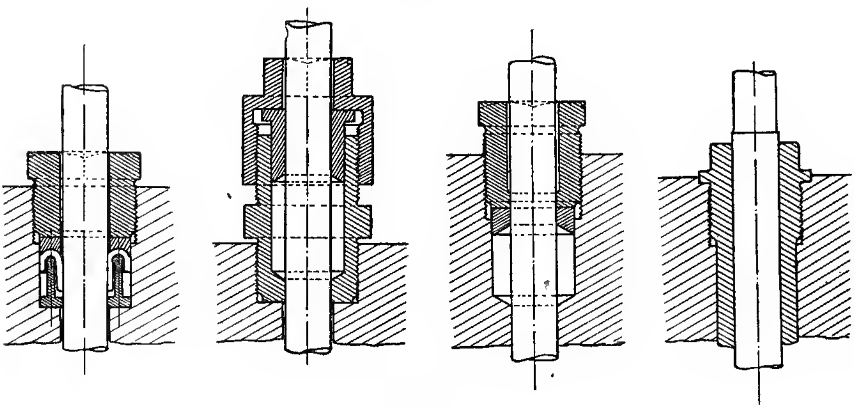


FIG. 166.

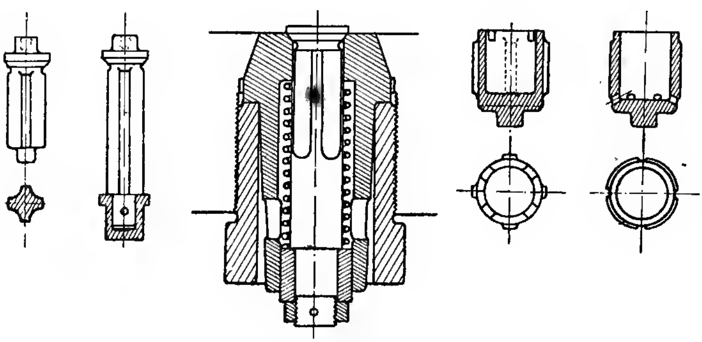


FIG. 167.

Estimated quantity of fuel per cycle $=\frac{0\cdot4\times250}{60\times60}=0\cdot0278$ lb.

Volume occupied by 1 lb. of fuel = about 31 cub. in.

Therefore volume of fuel per cycle $=0\cdot0278\times31=0\cdot86$ in.^{3}

Adding 50% to allow for overload, possible increase of fuel consumption, leakage of plunger, etc. :—

Stroke volume of plunger $=1\cdot5\times0\cdot86=1\cdot29$ in.^{3}

Which is satisfied by a plunger diameter of $\frac{3}{4}$ in. $\times$ 3 in. stroke.

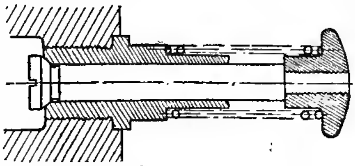


FIG. 168.

This size of plunger would only be permissible on a marine engine. If the cylinder belonged to a governed engine the stroke volume of the fuel pump plunger would need to be about four times the above figure, as it is found that good governing at all loads is only to be obtained by using about the last quarter of the stroke. This is probably due to the fact that the quantity of fuel consumed by the engine in a given time is not proportional to the load but more nearly proportional to the load plus a constant representing the engine friction. The actual position taken up by the governor and the effective stroke of the pump plunger at any specified proportion of full load are not easy to determine experimentally with great accuracy, but the angular positions indicated in Fig. 169, with reference to the fuel pump eccentric circle, are those generally used as the basis of calculation.

When one plunger is used to supply several cylinders the length of effective discharge period is limited by the condition that the latter should not overlap the injection periods. In estimating the capacity of a fuel pump driven off a vertical shaft the speed of the latter must be kept in mind, being usually the same speed as the engine, and in some cases 50% more.

The valves, hand plungers, etc., are suitable subjects for distributive standardisation. For example, a suction valve $\frac{5}{8}$ in. in diameter would be quite suitable for all sizes of cylinder (assuming one plunger per cylinder) up to about 20 in. bore provided that the use of fuels of exceptional viscosity were not

contemplated. For oils like crude Mexican, of the consistency (when cold) of tar, larger valves are probably advisable. With the valve arrangements in common use the diameter of the delivery valve is determined by that of the head of the suction valve plus adequate clearance for the flow of the oil round the latter.

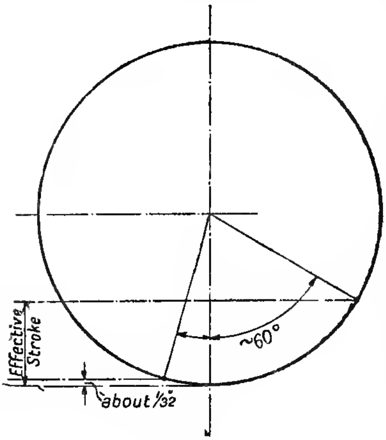


FIG. 169.

The general thickness of metal of the pump body is usually kept as uniform as possible to facilitate casting, and the nominal stress in the neighbourhood of the pump chamber based on a blast pressure of 1000 lb. per sq. in. is about 2500 lb. per sq. in. The design of a fuel pump affords ample scope for a draftsman's skill in many directions, in which numerical calculation plays a very small part, and the following suggestions are offered :—

(1) The arrangement generally to be neat and substantial and presenting an external appearance in keeping with its surroundings.

(2) The flanges and brackets by which it is secured to the framework of the engine to be unobtrusive and to have the appearance of growing as naturally as possible out of the main body of the casing, so as to convey an impression of rigidity and equilibrium.

(3) Every detail to be carefully studied, both with regard to its special function and also to economy in manufacture, efficiency always having precedence over economy. In particular, case-hardening and bushing of parts subject to wear must not be stinted, and provision should be made for lubrication of all moving parts.

(4) Valves and other internal mechanism to be easily accessible for inspection and overhaul.

(5) Arrangements to be made to catch all drips, both of fuel and lubricating oil, avoiding the use of trumpery sheet-iron guards and the like.

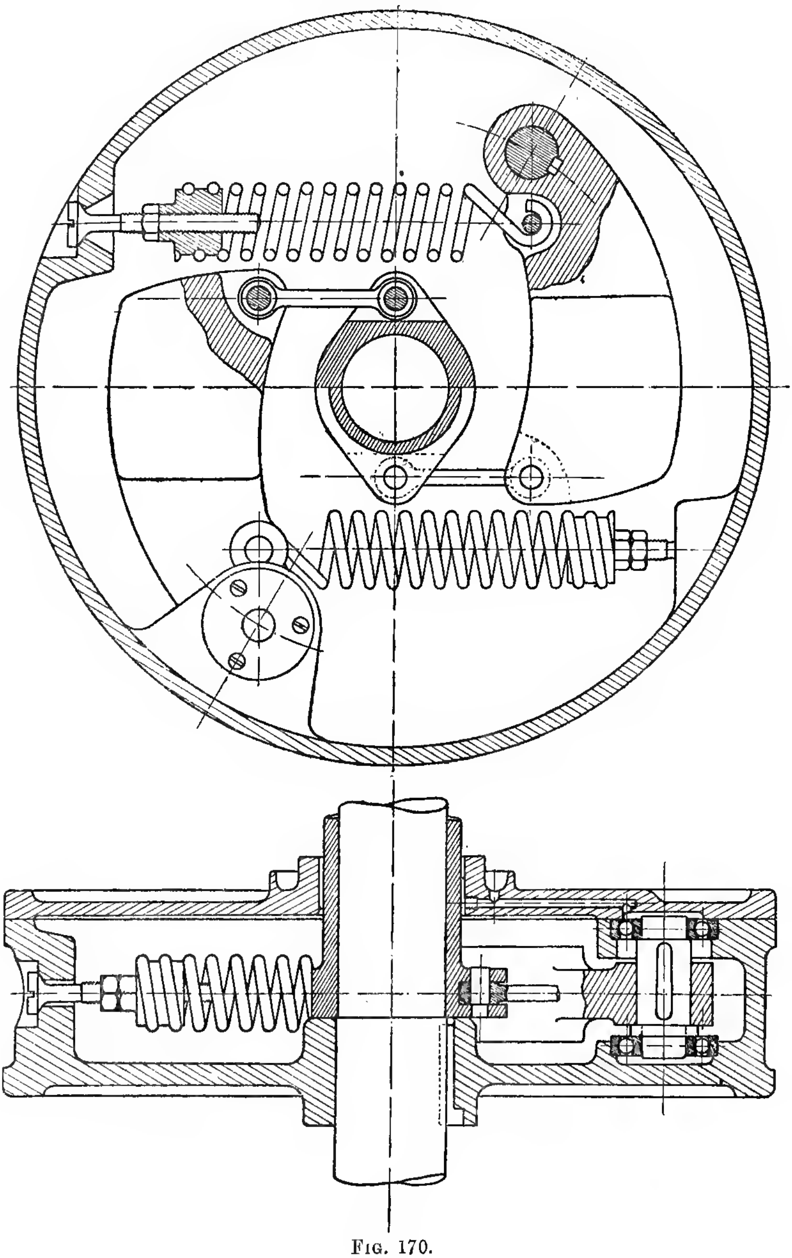


FIG. 170.

Many of the above principles apply of course to the design of any part of any high-grade machine, and they are mentioned here because the matter on hand provides an excellent opportunity of emphasizing them in a particular case, in which the subject is singularly free from the complications arising from calculations. When the discussion is transferred to some large part of a machine, in which the stresses are approximately determinate and the scope of the design appears to be limited by adjacent parts, it becomes increasingly difficult to reconcile the ideals of high-class design with the requirements of efficiency and economy and the skill of a designer may be gauged by the extent to which this difficulty is overcome. From this point of view no part of a design can be said to be finally determined until the whole design is complete, as there is always the possibility that a design for a certain part, perfect in itself, may require to be modified subsequently on account of its relationship, perhaps remote, to some other part as yet undetermined.

Governors.—It is not proposed to deal here with governor design generally, as that is a subject for a specialist in this particular department of mechanical design, but only to illustrate the application of governors to stationary Diesel Engines by means of a few examples, and to give the main lines of calculation for the type of governor shewn in Fig. 170, which is a type not usually standardised by governor specialists. The action of the weights in causing rotation of the central sleeve will be immediately obvious from the figure. The amount of this rotation between the limits of no load and full load should correspond with the angle $\sim 60°$ of Fig. 169, but as a safety precaution it is advisable to give the governor sufficient range to give a complete cut-out, and the sleeve should therefore be free to describe an angle of about 70°. The first stage in the design of the governor is to rough out a drawing similar to Fig. 170, fulfilling all the requirements as to space, accessibility, etc., and in which the above angular rotation is secured. As regards the size of the governor, it is generally wise to avail oneself of all the space obtainable. The next step is to find the mass and centre of gravity of the weights and the positions of the latter in the extreme in-and-out positions. A diagram similar to Fig. 171 should now be constructed, in which the abscissæ are distances of the weight from the centre of the shaft in inches and the ordinates are the centrifugal forces at

these distances at no load speed and full load speed respectively. Point "P" corresponds to "no load" speed and "no load" distance from centre, and point "Q" the same quantities for full load. The line PQ then determines the properties which the controlling spring would have to possess if it were connected to the weight at its centre of gravity. These properties are as follows :—

(1) The initial tension, when the weights are full in, is equal to the centrifugal force corresponding to the point "Q."

(2) The weight of the spring per inch extension is equal to the slope of the line PQ, that is the amount in lb. by which the ordinate increases as the abscissa increases by one inch.

Actually the spring is attached to the weight at a point nearer to the fulcrum than the centre of gravity, and both the initial tension and the rate as found thus require to be increased in the ratio $\frac{k}{l}$, where :—

k = Distance of the weight fulcrum from the line joining the C.G. of the weight to the centre of the governor.

l = Distance of the weight fulcrum from the line of action of the spring.

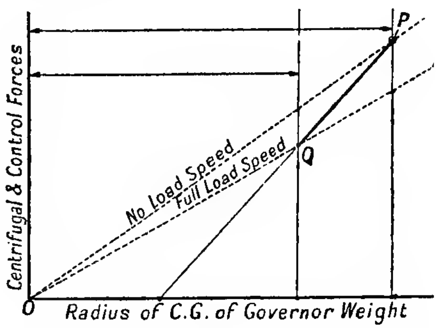


Fig. 171.

This very simple construction, repeated as often as may be necessary in the process of trial and error, contains all the dynamical calculation required to ensure sensibility and stability, but it is advisable to provide adjustments for spring tension, in the manner shewn in the figure, to allow for un-

avoidable errors and routine adjustment on the test-bed. Strictly speaking, the diagram shewn in Fig. 171 should be corrected to allow for the versed sine of the arcs described by the points of suspension, and so on; but these are practically negligible. Other types of governor are designed on similar lines, but are usually complicated by link mechanism, of which the variations of configuration are not negligible.

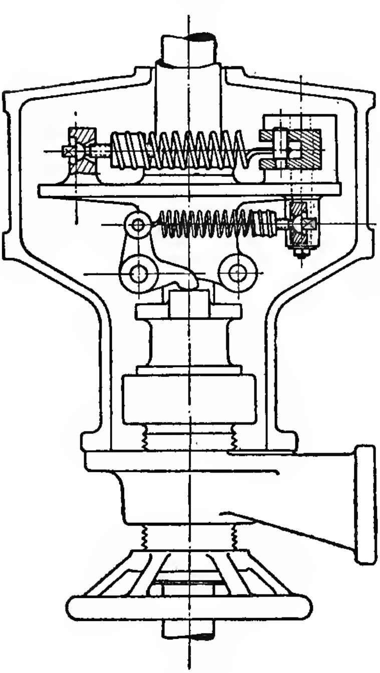


FIG. 172.

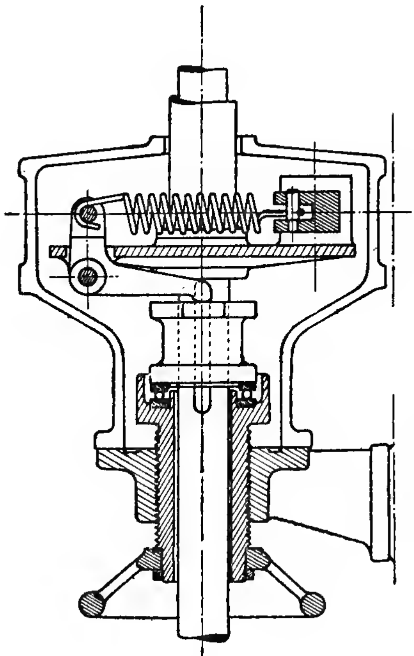


FIG. 173.

Variation of speed during the running of the engine is readily secured by transferring a part of the controlling force to an auxiliary spring, the tension of which can be varied by mechanism provided for the purpose, as shewn in Fig. 172, or by varying the tension of the main spring itself, as in Fig. 173.

Some points to be observed in governor design are :—

(1) Weights as heavy as possible, to give power and consequently render the effects of friction negligible.

(2) Springs to be readily adjustable.
(3) Small pin-links, etc., to be as substantial as conveniently possible.
(4) Friction to be reduced to a minimum by ball-bearings.
(5) Joints other than ball-bearings to be bushed and provided with well-hardened pins.
(6) Lubrication, both as regards supply of lubricant to the working parts and systematic disposal of the surplus, to be considered carefully.

The general disposition of the governor and fuel pump with respect to the framework of the engine is shewn in Figs. 174,

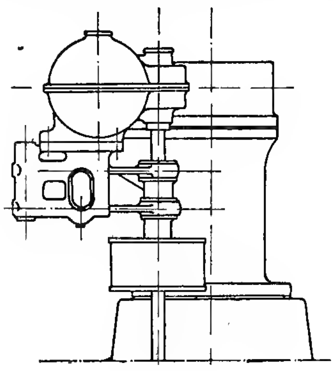


Fig. 174.

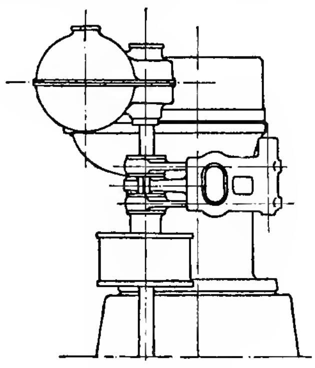


Fig. 175.

175 and 176, in three cases. In Fig. 174 the governor is of the angular movement type described above and illustrated in Fig. 170, driven off the vertical shaft from which the cam-shaft receives its motion. The fuel pump is of the horizontal plunger type, receiving its motion from eccentrics mounted on the same vertical shaft. The fuel pump body is supported by a facing on the lower side of the case containing the upper spiral gears.

In the arrangement shewn in Fig. 175 the fuel pump is attached to the cylinder jacket, but in other respects the details are similar to those of Fig. 174.

The governor shewn in Fig. 176 is of the more usual type characterised by a sleeve which is mounted on a feather and which rises as the engine's speed increases. The pump is of the vertical multi-plunger type, and regulation is effected by

rotation of an eccentric shaft, on which are hinged the levers which operate the auxiliary plungers.

The arrangements described briefly cover the bulk of the

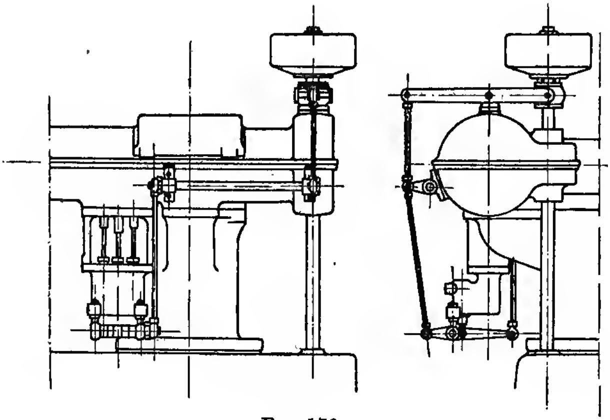


FIG. 176.

fuel pump and governor mechanisms found in practice, but mention must be made of some modern refinements which are coming increasingly to the fore.

(1) Control of fuel valve opening. At light loads the duration of opening of the fuel valve is greater than necessary if uncontrolled and the instant of opening which is most favourable for full load running is inclined to be late for light load running. At least one firm has attacked this problem of governor control of the fuel valve operating mechanism.

(2) Blast pressure control. This question is closely allied with (1), as a shortened opening period would lead to increase of the blast pressure if the latter were not corrected. Apart from this the blast pressure has in any case to be altered in accordance with the load (unless cylinders are cut out of operation) if good combustion is to be secured at all loads, including no load. The blast pressure is placed under the control of the governor by means of a throttle slide on the compressor suction.

(3) Pilot ignition. This refers to cases where exceptionally refractory oils are being used which require for their

combustion a preliminary charge of a lighter oil, such as Texas oil, which is deposited in the pulveriser in advance of the main charge by a small auxiliary pump provided for the purpose. The necessity for this device appears likely to be obviated by improvements in fuel valves and the flame plates in particular.

Fuel Injection Valves.—It now remains to deal with the valve by means of which the fuel is injected into the combustion space and to which oil is delivered by the fuel pump for this purpose. These may be broadly classified as the open and closed types respectively, and as the former form a relatively small class at present it is convenient to dispose of them first.

Open Type Fuel Valves.—Fig. 177 is a diagrammatic view of such a valve, omitting all detail not required to illustrate

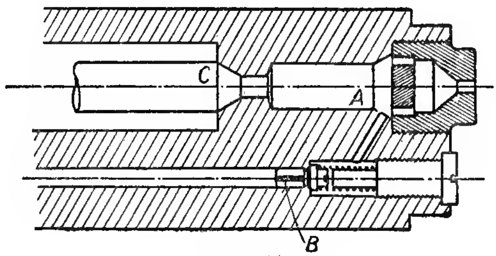


Fig. 177.

the bare principle. Oil is delivered to the space A past the non-return valve B by means of the fuel pump, and this type of fuel valve derives its name from the fact that the space A is in constant communication with the interior of the cylinder. It is to be noticed that the fuel pump is not required to deliver against the pressure of the blast air as the latter is restrained by valve C. The latter is opened by appropriate gear at the predetermined instant for injection and carries with it the fuel oil contained in the space A. The action appears to be highly efficient in pulverising effect and excellent fuel consumptions have been reported for engines in which these valves have been fitted. This type of fuel valve appears to have been devised in the first instance for use in horizontal engines in which it was anticipated that the more usual type of fuel valve would be at a disadvantage. A valve working on a somewhat similar principle has been tried, from all accounts successfully, on

vertical engines, but has not yet, to the author's knowledge, become a standardised fitting.

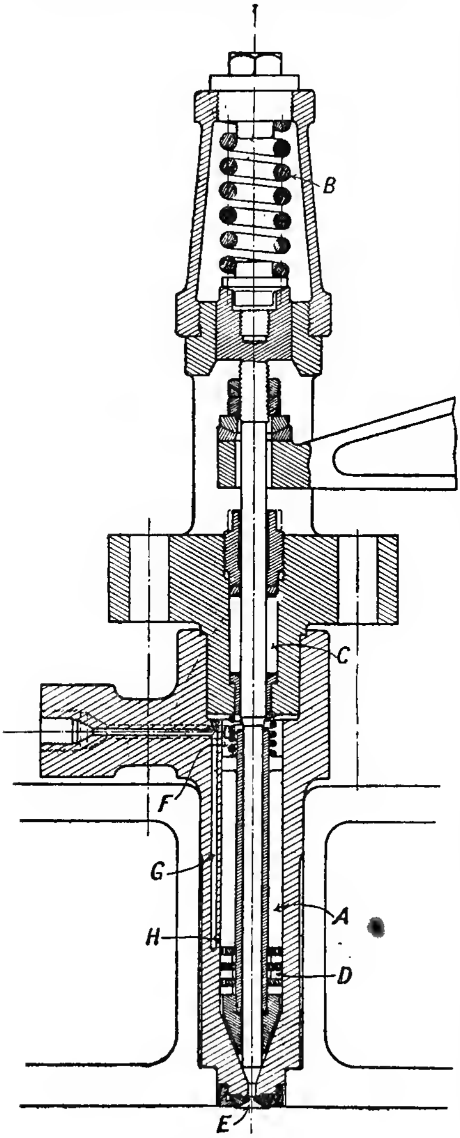


Fig. 178.

Closed Type of Fuel Valves.—In this type communication between the combustion space and the interior of the fuel valve only exists during the injection period, when the flow is always in the same direction, apart from such derangements as stuck valves or failure of the blast pressure.

Fig. 178 shews what may not improperly be called the Augsburg type of fuel valve. Apart from the cast-iron body, the construction of which is sufficiently illustrated by the drawing, the principal parts are :—

(1) Needle valve A.
(2) Spring B.
(3) Stuffing box C.
(4) Pulveriser D.
(5) Flame plate E.

The needle valve is usually made of special steel, case-hardened in way of the stuffing box to prevent cutting by the packing. Accurate alignment of all parts of the needle is essential and readily secured by grinding between fixed centres. The lower part of the needle is preferably reduced in diameter by a

few thousandths, as a certain temperature gradient exists between the needle and the pulveriser tube which may lead to seizure if sufficient clearance is not allowed. The tip generally has an angle of about 40 degrees. The needle spring, in addition to returning the needle to its seat against the pressure of the blast air, has to deal with the friction of the stuffing box, and may be figured out on the basis of a pressure of 1500 lb. per sq. in. over the sectional area of the needle at the stuffing box. The latter is usually provided with a screwed gland.

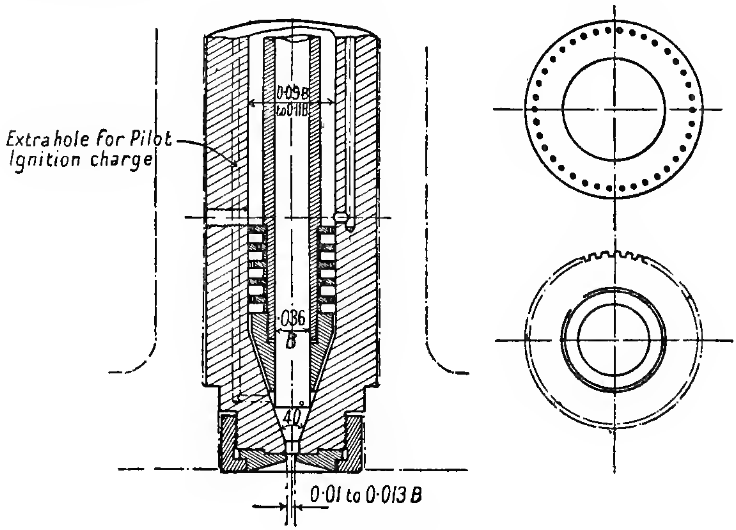


Fig. 179.

The pulveriser tube is held on its seat by a stiff spring, and serves the double purpose of affording some support to the needle and retaining in their relative positions the rings and the cone which play an important part in pulverising the fuel. It will be clear from the figure that the pulveriser is surrounded by blast air, which enters at F. The fuel is introduced by means of a narrow hole G, at a point H immediately above the top ring. If the point H is located too high the oil fails to distribute itself evenly round the pulveriser rings and inefficient combustion results.

Fig. 179 shews the injection end of the pulveriser, together

with the flame-plate and nut, to a larger scale. The details shewn are those in most common use, but are subject to variation in the practices of different manufacturers. The proportions shewn are roughly indicative of good practice, but it must be admitted that the rule of linear proportionality does not appear to be rational in this case. Experience in this matter discloses two facts :—

(1) That for a given engine there is a certain minimum diameter of pulveriser ring, below which results are not satisfactory (about 9% of the cylinder bore).
(2) That as cylinders are increased in size it becomes increasingly difficult to obtain a high M.I.P.

These suggest the following hypothesis :—

That the best results are to be obtained when the depth of oil in the pulveriser before injection is a certain amount, and the same for cylinders of all sizes. If this is true, then the area of the pulveriser ring should be in proportion to the cylinder volume. This would lead to the diameter of pulveriser rings being made proportional to the cylinder bore raised to the power of 1·5. Such a rule has not been adopted, and would probably lead to inconveniently large valves in the larger sizes of engines, but the question would appear to offer some inducement to research. A very large number of different types of pulveriser are in use, and have been described in the technical press ; but it still remains to be proved that they are more efficient than the common variety shewn in Fig. 179. A neat form of pulveriser tube, which dispenses with the long narrow hole drilled in the fuel valve casting, is shewn in Fig. 180, from which it will be seen that the oil is led to an annular space A at the top of the tube, whence it flows downwards to the pulveriser rings via a number of grooves in the surface of the pulveriser tube. Holes B are provided to give passage for the blast air.

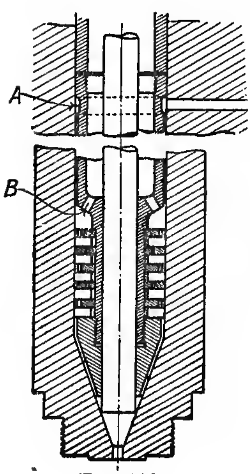


FIG. 180.

Swedish Type of Fuel Valve.—Fig. 181 shews the construction of this type of valve, which has also been widely adopted and which is characterised by the fact that

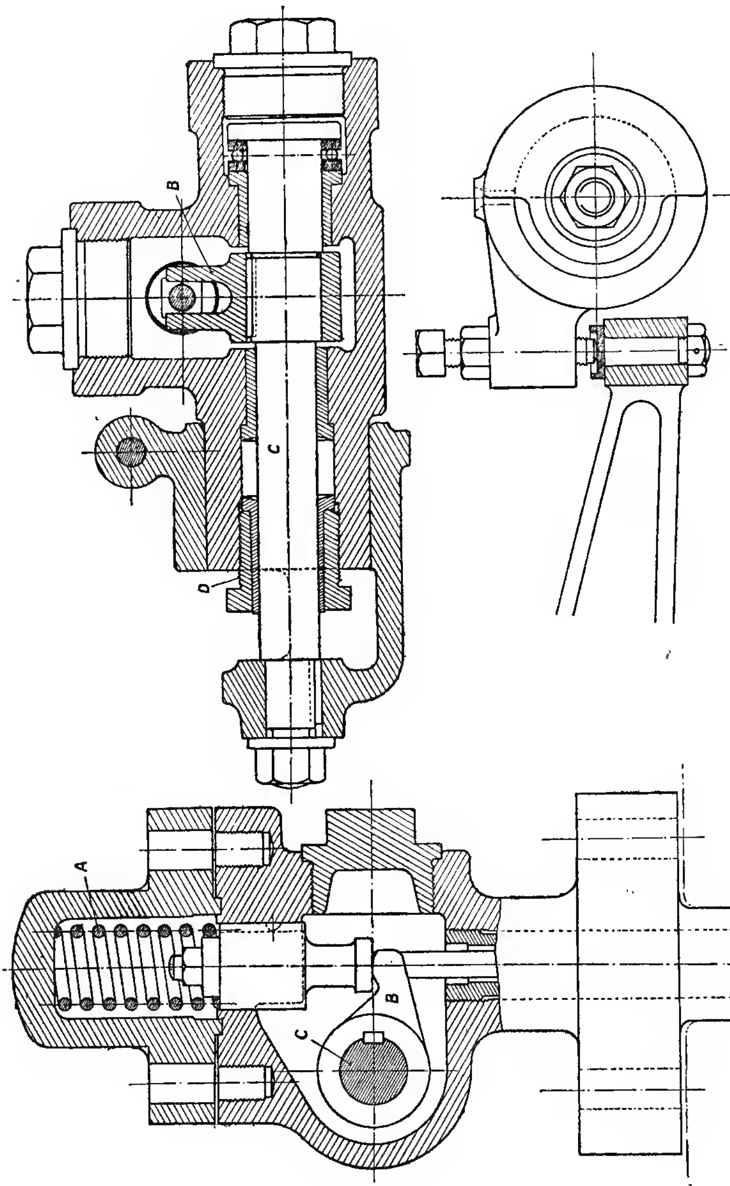


FIG. 181.

the needle is completely enclosed within the casing and is subject on all sides except the extreme tip to the pressure of the blast air. On this account the spring A does not require to be as strong as that of an Augsburg type of valve of the same size. The needle is lifted in working by the lever B attached to a cross-shaft C, the end of which penetrates the casing through a stuffing box D. The mechanical means by which end thrust on the cross-shaft and bending actions on the overhung end, due to the pressure on the external lever, are dealt with, will be clear from the figure without further explanation. The use of this type of valve appears to be limited at present to those designs in which the requirements of other parts of the valve gear necessitate the fuel valve operating lever being arranged off the centre line of the cylinder cover.

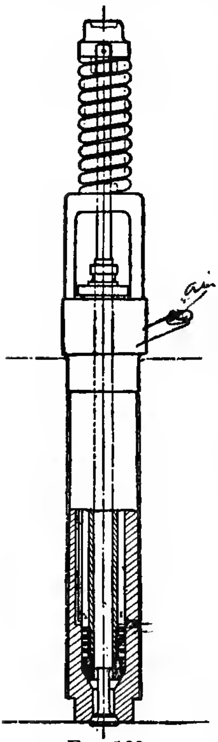


FIG. 182.

Burmeister Fuel Valve.—The construction of this valve is shewn diagrammatically in Fig. 182, and its outstanding features are the use of a mushroom valve, the extreme simplicity of the whole arrangement, and the fact that the valve is opened by a downward movement. The latter is a particularly valuable feature as it secures uniformity of valve gear and ease of withdrawal.

The four classes of fuel valve described above include as members practically all the fuel valves in use on Diesel Engines at present. Each type has its advantages, but no one of them can be said to hold the field. Something similar might be said for the enormous variety of pulverisers patented and in actual use. It seems doubtful if any of these can claim outstanding efficiency. When pilot injection of a less refractory oil is used to facilitate the use of tar oil as fuel an additional hole has to be drilled in the fuel valve, as shewn dotted in Fig. 179. The question of burning tar oil is still in the experimental stage in this country, but the results so far obtained hold out hopes that it will be possible to dispense with pilot ignition in favour of special arrangements of a simpler character in connection with the fuel valve details.

Some of the arrangements by means of which fuel valves are operated are shewn in Figs. 183, 184, 185 and 186. The long lever, which is a feature of all these schemes, is usually of cast steel, and should be of stiff construction. The fulcrum on which the lever hinges is common to the levers which operate

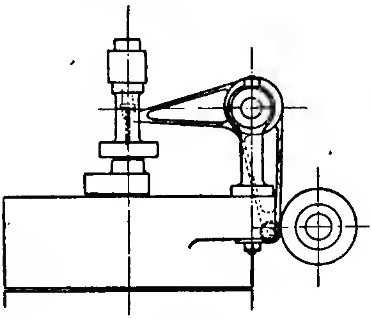


FIG. 183.

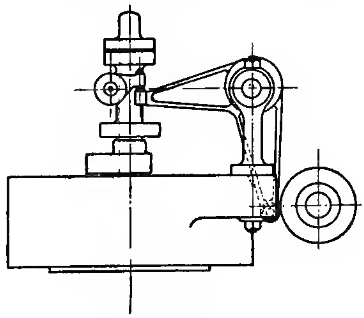


FIG. 184.

the other valves, viz. air and exhaust valves in the case of four stroke engines and scavenge valves in the case of two stroke engines, and starting valves in both cases. With land engines, and many marine engines, it is usual to mount the fuel and starting valve levers on eccentric bushes mounted on the

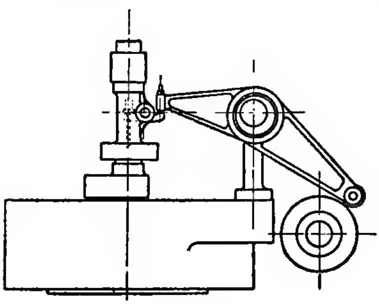


FIG. 185.

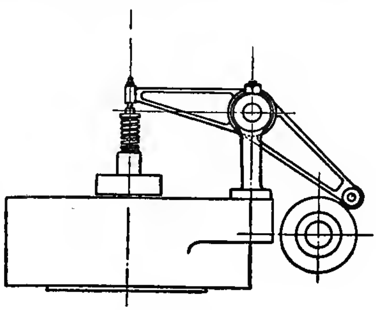


FIG. 186.

fulcrum shaft at such angles that the operation of putting the starting valve into gear automatically puts the fuel valve out of gear and vice versa. This is considered in detail in Chapter XI.

The use of the needle type of fuel valve in conjunction with a single lever necessitates the latter being so disposed that its

roller is rendered more or less inaccessible by the cam-shaft, particularly if the latter runs in a trough (see Fig. 184). The difficulty may be got over by providing a small intermediate lever, as shewn in Fig. 185, to reverse the direction of motion. In spite of the objections which have been raised against this arrangement it appears to be satisfactory in practice.

Design of Fuel Valves.—An approximate rule for the internal diameter of the body has already been given, being the same as the diameter of the pulveriser rings. The thickness of the walls (cast iron) may be from a third in large valves to a half in the case of small valves of the internal diameter. If the valve is of the Swedish type this thickness will be approximately constant throughout the body of the valve, except in the neighbourhood of flanges, etc. If of the Augsburg type, those parts of the body not subject to pressure may be a little thinner. In all cases a good rigid job should be aimed at, as lack of alignment leads to sticking of the valve. The pulveriser tube is made of steel and the details, such as rings and cones, of steel or cast iron. The Swedish type of valve requires special care to be devoted to the design of the cross-shaft and its fittings, in order to obtain freedom under load, adequate bearing surface and accessibility of the stuffing box. As regards the valve as a whole, the designer should aim at shapely solidity and avoid flimsiness of detail.

With the Augsburg type of valve (Fig. 178) the load necessary to lift the needle is the spring load less the product of the blast pressure and the area of the needle at the stuffing box (approx.) plus the gland friction. With the Swedish type (Fig. 181) the load may be taken as approximately equal to the spring pressure plus the product of the blast pressure and the area of the needle at its seat. This load evidently induces bending and twisting actions, which the cross-shaft should be proportioned to carry with a low stress. The weakest section is generally at the reduced diameter to which the external lever is keyed. The key itself should be amply proportioned, and is preferably made of tool steel. The ball thrust must be proportioned to the load obtained by the product of the maximum blast pressure into the sectional area of the cross-shaft at the stuffing box. The flame plate is of nickel steel and the diameter of the hole is usually about 1% of the cylinder bore, but the best size for any particular case must be found by experiment. The flame plate nut may be of steel or bronze secured

to the fuel valve body by a fine thread and provided with flats to accommodate a spanner.

The main points in the design of fuel valves may be summarised as follows :—

(1) Rigidity and alignment of casing.
(2) Alignment of the needle and its guide.
(3) Freedom of all working parts.
(4) Sturdy proportions for all small details.

Literature.—Renolach, N. O., "Tar Oils as Fuel for Diesel Engines."—*Internal Combustion Engineering*, July 15th, 1914, *et seq.*

Porter, G., "Tar Oil Fuel and Diesel Engines."—Diesel Engine Users' Assoc., May 24th, 1917.

Smith, P. H., "Two Essential Conditions for Burning Tar Oil in Diesel Engines."—Diesel Engine Users' Assoc., May 16th, 1918.

CHAPTER XI

AIR AND EXHAUST SYSTEM

Four Stroke Engines.—So far as four stroke engines are concerned the parts included in this system are :—

The air suction pipe.
Air suction valve.
Exhaust valve.
Exhaust piping.
Silencer.

In the neighbourhood of cement works, or other sources of grit, a suction filter is sometimes added, with a view to preventing foreign matter from reaching the interior of the cylinder along with the indrawn charges of air.

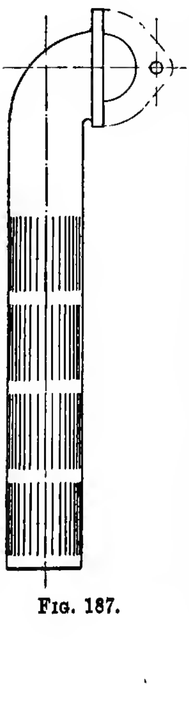


FIG. 187.

Air Suction Pipes.—The usual form of suction pipe for both land and marine engines is shewn in Fig. 187, and is intended to serve the double purpose of muffling the sound caused by the rush of air past the inlet valve on the suction stroke and also to prevent in some measure the ingress of dust. To be of real service in either capacity the slots should not exceed about 40/1000 of an inch in width, and in use must be kept clear of dirt, otherwise the volumetric efficiency, and consequently the power of the engine, will be seriously impaired. To be reasonably effective as a silencer the slots should cease within a foot or so of the point where the pipe joins the cylinder cover. To prevent throttling the aggregate area through the slots is usually made about 1·5 to twice the clear area of the pipe. Assuming the slots are spaced $\frac{3}{8}''$ apart, the slotted length of the pipe works out to about $3\frac{1}{2}$ to $4\frac{1}{2}$ times the bore. In very

expert foundries these slots are formed in the casting (cast iron). Where the slots have to be milled aluminium is a convenient but expensive material for this purpose. Welded tubes of sheet iron are sometimes used, but are liable to become dented and unsightly.

An efficient and durable arrangement, shewn in Fig. 188, consists of a common collecting pipe in communication with all the cylinders and ending in a trumpet-shaped piece which is very effective in muffling the sounds of suction. The trumpet fitting is similar to a cornet mute, and consists of internal and external members so arranged that the space between them has a sectional area increasing outwards whilst the distance between the two members diminishes.

The noise question is most effectively disposed of by carrying the suction pipe outside the building in the case of a land engine or on deck in the case of a marine engine. If an air filter be added the arrangement is ideal but expensive.

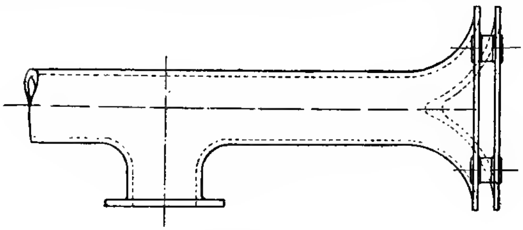


FIG. 188.

Where a separate suction pipe, as in Fig. 187, is fitted to each cylinder the bore is usually made equal to that of the suction valve or slightly larger. Where a common suction pipe is provided the bore may be anything up to about double this size, according to the number of cylinders to be supplied and the length of the pipe.

Suction Valves.—A typical suction valve is shewn in Fig. 189. The valve proper may be of carbon or nickel steel, in the form of a drop forging. The upper end is guided by a little piston, which also takes the thrust of the spring. In the example shewn the guide piston is of chilled cast iron extended in the form of a nut and provided with a cup-shaped cavity to accommodate the operating tappet. The casing being in two parts, it is necessary to finish each part on a mandril to ensure alignment in position. It is not necessary at this point to describe

variations in the details of suction valves, as the latter are usually made similar to the exhaust valves (which will be described in detail later) except for such special features as are necessary to deal with the heat effects to which the latter are subject.

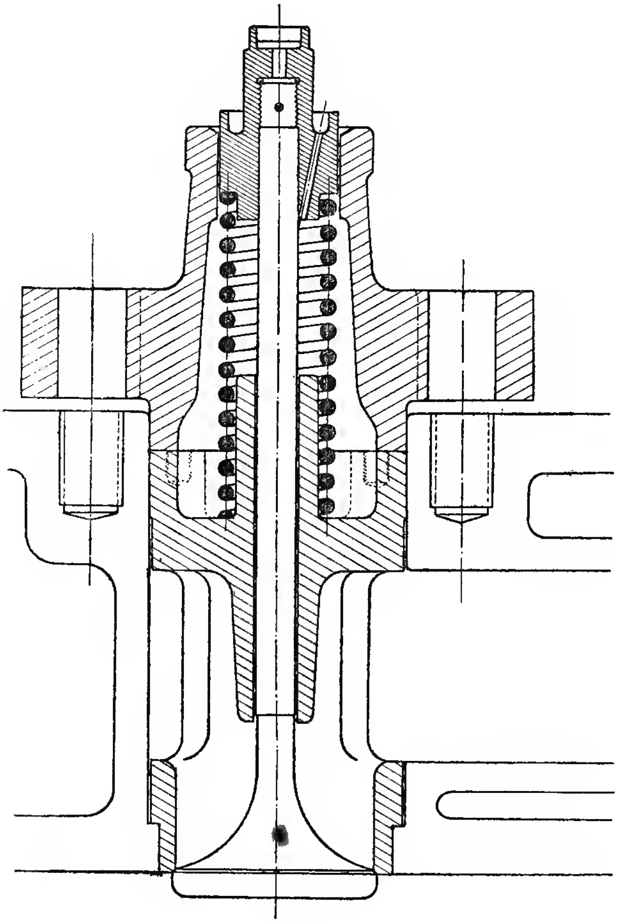


FIG. 189.

Dimensions of Air Suction Valves.—The requisite diameter for an air suction valve may be regarded as determined by the mean vacuum allowable on the suction stroke. Taking this to

be 0·6 lb. below atmospheric pressure, the theoretical mean velocity is found from Fig. 13 to be about 280 feet per second, and taking the mean coefficient of discharge to be 0·70 this gives a mean apparent velocity of 195 feet per second. Now as regards the mean opening area of the valve, if we assume a harmonic cam opening and closing exactly at the upper and lower dead centres respectively, then the mean opening would be just half the maximum opening if the maximum lift is made $=\frac{1}{4}$ of the value diameter. Actually the valve is always arranged to open before top centre and close after top dead centre, so that the mean area is usually more like 0·6 of the maximum area.

Adopting this figure, we may write :—

$$195 = V_a = V_p \times \frac{0{\cdot}785\,B^2}{0{\cdot}65 \times 0{\cdot}785\,d^2}$$

Where V_a = Apparent mean velocity of air in feet per second.
V_p = Mean piston speed in feet /seconds.
B = Bore of cylinder.
d = Diameter of suction valve.

From which,

$$\frac{d}{B} = \sqrt{1{\cdot}54\,\frac{V_p}{195}} = \sqrt{\frac{V_p}{127}}$$

Values of $\frac{d}{B}$ calculated from this formula for various piston speeds are given below and agree well with average practice.

Piston speed in ft. per min. .	500	600	700	800	900	1000	1100	1200
Ratio d÷B .	·257	·281	·303	·324	·343	·362	·380	·397

In the best practice the maximum lifts of the air and exhaust valves are frequently made as much as 0·35 of the valve diameter, as although the extra lift does not increase the maximum available area the mean area is increased and the opening at the dead centre is augmented without adopting an unduly long opening period or an awkward shape of cam.

The air and exhaust valves are usually operated by the lever arrangement shewn in Fig. 186 with reference to fuel valves.

Exhaust Valves.—In some small engines the exhaust valves are similar to and interchangeable with the air suction valves, but with medium and large size engines, owing to the larger dimensions of the exhaust valves, and in consequence their slower rate of cooling by conduction to the surrounding media,

special arrangements have to be made to conduct heat away from the valve seat which would otherwise become pitted and grooved in a short time.

With cylinders up to about 16 in. bore the trouble can be reduced to reasonable proportions by providing the valves with cast iron heads, which are less liable to become pitted than those of nickel steel. Ventilation of the casing by means of perforations is also found useful. Some types of cast-iron exhaust valve heads which have proved successful in practice are shewn in Fig. 190. With cylinders much in excess of 16 in. bore some form of water-cooled casing is desirable, and the simple arrangement shewn in Fig. 191 has proved most

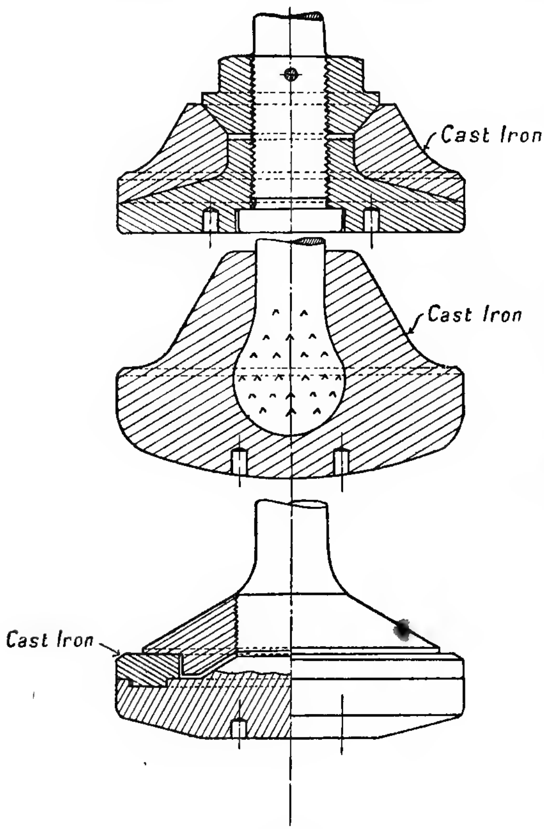


FIG. 190.

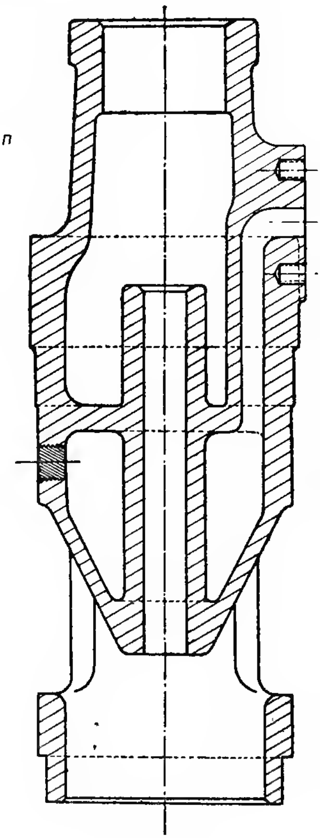


FIG. 191.

effective with the very largest cylinders without resorting to the expedient of cooling the valve itself by direct means. It appears that the flow of heat from the seating to the water-jacket is sufficient to keep the temperature of the former at a suitable low value so long as the mean indicated pressure does not exceed a reasonable figure dictated by other considerations.

The valve spindle is liable to become stuck by carbonaceous deposit, and should therefore be about 20/1000 slack in the casing. Furthermore, arrangements should be made to feed a little paraffin to this point from time to time.

The upper or guide end requires some care in detailing,

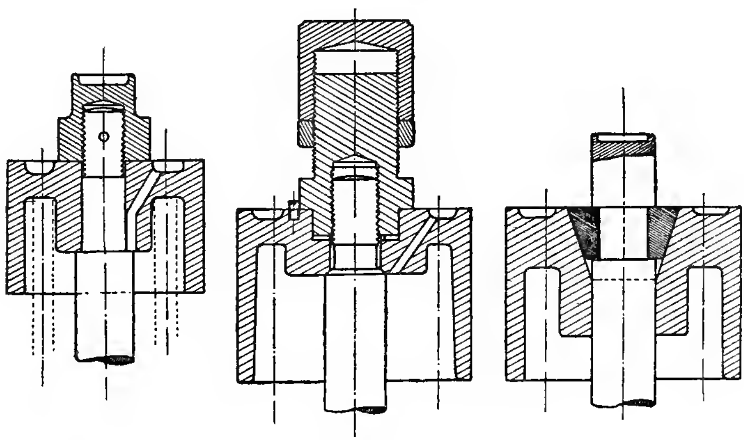


FIG. 192.

particularly in high speed engines, to prevent the nut from slacking back or the threads becoming stripped. The arrangement shown in Fig. 189 is simple and effective; other arrangements are shown in Fig. 192 for slow speed engines.

Valve Casings.—Two types of valve casings have been illustrated in Figs. 189 and 191, and further modifications are shown in Fig. 193. The valve seating is made in a separate piece, which is readily replaceable when worn by an interchangeable spare. After repeated regrindings a ridge is formed which has to be turned off, a process more conveniently carried out on a light ring than on a whole casing. If no loose seat is provided it becomes necessary in time to turn down the under surface of the casing flange, in order to bring the valve seat

flush with the under side of the cover. The port in the casing provided for the flow of exhaust gas should accurately coincide with the corresponding port in the cover, and it is usual to provide an equal and opposite dummy port to preserve the symmetry of the casting and diminish chances of distortion due to expansion or growth. The depth of the metal under the ports should not be reduced to too fine a limit, otherwise there is danger of leakage on the compression and expansion strokes, due to deflection or even cracking of the metal at this point. The number of studs (two to four) used to secure the casing to the cylinder cover is purely a matter of convenience in arranging the gear on the cover or cylinder head. Two studs properly proportioned are sufficient for the largest valves, given an adequate depth of flange and good connection of the latter to the body.

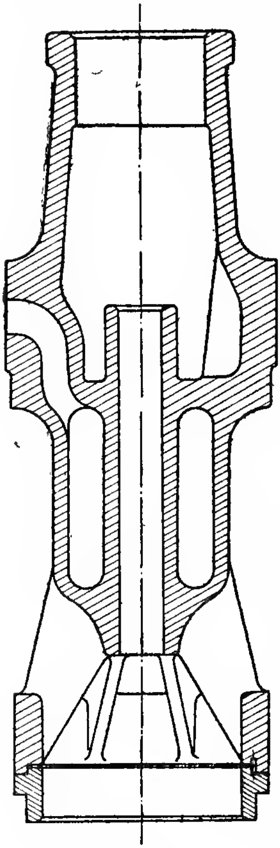


Fig. 193.

Exhaust Lifting Devices.—Some form of exhaust lift for breaking compression is usually fitted to both land and marine engines. With the former the use of such a device on shutting down the engine obviates the tendency of the engine to swing in the reverse direction to that for which it was designed, just before stopping, and also facilitates turning the engine by hand. With marine engines some such device should come into operation automatically on reversing to prevent the possibility of compressing an unexhausted charge when motion is begun in the reverse direction. An expansion stroke executed in the ahead direction becomes a compression stroke in the astern direction and vice versa. Two hand devices are shown in Fig. 194. Type A consists of a substantial steel lever provided with a slotted end which normally keeps it in a vertical position clear of the gear. To bring the lever into operation it

is lifted to the extent of the slot and allowed to fall forwards. A projection on the lever then slips under an extension of the roller-pin on the first occasion of the valve being lifted and prevents its return. Type B consists of a link and screw, by means of which the valve may be depressed during the running of the engine and which normally lies alongside the casing. Fig. 195 shows three arrangements suitable for marine engines. In type A the valve is depressed by a pneumatic cylinder arranged above

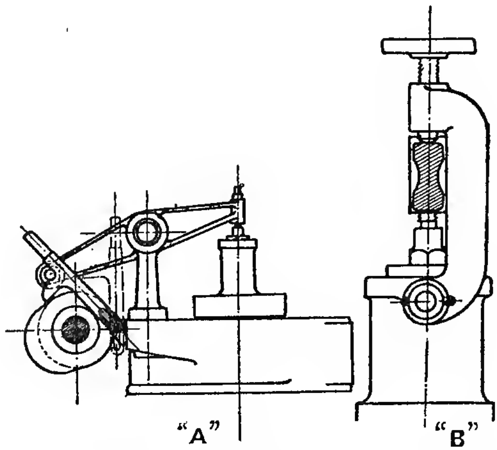


FIG. 194.

and in line with the valve. Type B consists of a series of cams (one cam over each exhaust valve lever) mounted on a shaft running from end to end of the engine. A turning movement of this shaft simultaneously depresses all the exhaust valves. Type C is appropriate to those engines in which the valve levers are operated by push rods. The first movement in reversing consists of swinging the lower end of the push rods out of the range of operation of the cams. By a scheme of linkwork, which is obvious from the illustration, the same movement introduces a lever with an inclined face under a roller provided for this purpose attached to the valve lever, with the result that the valve is held off its seat until the push-rods regain their normal working positions.

Proportions of Exhaust Valves and Casings.—The diameter of the exhaust valve is almost invariably made the same as that of the air valve, in order to simplify manufacture and give

a symmetrical arrangement of valve gear. On the exhaust stroke the area through the valve is more than sufficient, but a fairly early opening (40 to 50 degrees before bottom dead centre) is necessary to effect a rapid fall of pressure. The dimensions of the various parts of the valve and casing are with a few exceptions matters of experience only, and the approximate proportions given below with reference to Fig. 196 are representative of average practice. The figures are expressed in terms of the valve diameter as unit.

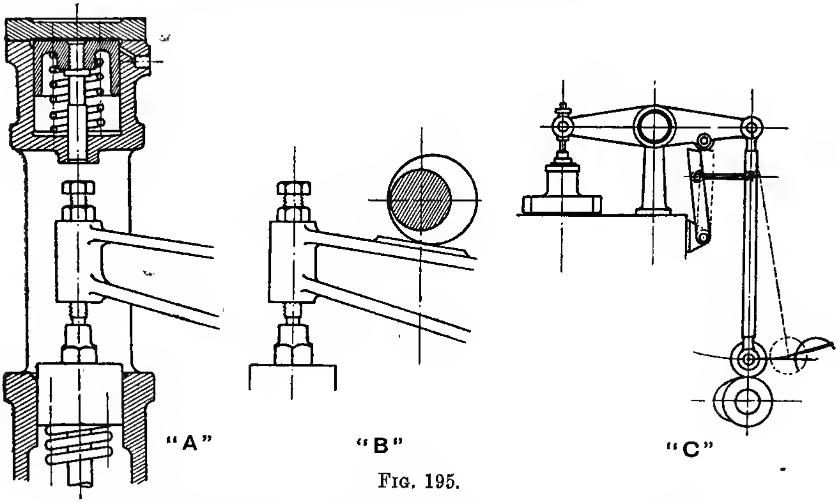


Fig. 195.

The larger figures refer generally to the smaller sizes of valve. The spring, which will be considered later, has usually between twelve and twenty-two turns. A large number of turns reduces the range of variation of stress, and consequently increases resistance to fatigue.

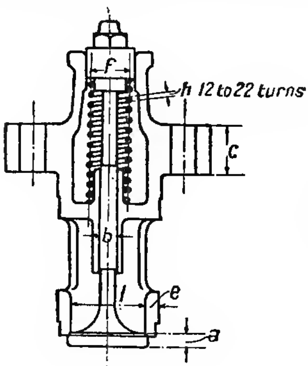
On the other hand, a spring with a small number of turns has less tendency to buckle.

The seating of the casing in the cylinder cover is almost always made square now instead of conical as formerly. The width of seating need not exceed a quarter of an inch in the largest sizes. The seating of the valve in the casing is usually made at an angle of 45 or 30 degrees, and the width of the seating varies greatly in different designs. Narrow seatings about an eighth of an inch in width appear to be least subject to pitting, and seatings as narrow as 1/25th of an inch have

been used successfully. For large marine engines seats about ⅜ inch wide seem to be preferred. The spindle clearance may be about 20/1000 and the guide piston clearance about 10/1000 in all sizes.

The size of the holding-down studs may be found from the standard table (page 130), the total load being based on a pressure of 500 lb. per sq. in. over the least area of the casing where it makes joint with the cylinder cover.

The casing lugs should be amply proportioned, particularly in cases where the castings are not above average quality, and they should have a good hold on the cylindrical part of the casing, reinforced if necessary by internal ribbing. It is not



a	*b*	*c*	*e*	*f*	*h*
0·15 to 0·18	0·20 to 0·24	0·45 to 0·56	0·07 to 0·17	0·46 to 0·64	0·06 to 0·08

FIG. 196.

unknown for these lugs to break off in tightening up the studs, so it is as well to err on the strong side.

Exhaust Valve Springs.—Apart from the weight of the valve in itself there are causes tending to open the exhaust valve when it should be shut, viz. :—

(1) The vacuum on the suction stroke.
(2) With four cylinder engines especially, the first rush of exhaust from the neighbouring cylinder if the latter discharges into the same collecting pipe.

In addition, the inertia of the valve and any levers, rods, etc., in connection therewith, tend to make the latter lose contact with the operating cam in the neighbourhood of maximum lift. These various influences are overcome by fitting a spring, the normal load of which is equivalent to a

pressure of about 10 lb. per sq. in. of valve area in the case of slow speed engines, and anything up to about 20 lb. per sq. in. or more in the case of high speed engines. A method of computing the inertia effect will be dealt with in some detail as there is a tendency for higher speeds to be used in practice, and the principles involved have a wide application in the design of high speed machinery generally.

Inertia Effect of Valves.—Consider the simple arrangement shown in Fig. 197, consisting of a valve directly operated by a cam without intermediate members. Let the weight of the valve guide and roller, etc., be "W" lb. The effect of the inertia of the spring may be allowed for by adding one-third of its weight to that of the other parts. Let "x" be the distance in inches of the valve from its seat at any instant. If the shape of the cam and the speed of the cam-shaft be known it is possible to express "x" in terms of the time "t" in seconds counted from the instant at which the valve begins to lift, by means either of an equation or a graph exhibiting the lift on a time base. If this equation (equation of motion) is available, then one differentiation with respect to "t" gives an expression for the velocity denoted by "$\dot{x}$," and a second differentiation gives the acceleration denoted by "$\ddot{x}$." If the relation between x and t is given by means of a graph, then the differentiation may be done by one or other of the graphical methods explained in books on practical mathematics. In either case, x being measured from the valve seat outwards, positive values of $\ddot{x}$ denote inertia effects tending to press the roller against the cam, and negative values of $\ddot{x}$ denote inertia effects tending to cause the roller to lose contact with the cam. Here we are only concerned with the negative values of $\ddot{x}$. If X denotes the resultant force on the valve, neglecting all effects except those due to the inertia,

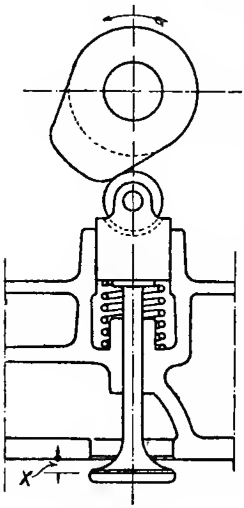


Fig. 197.

Then, $$X = \frac{W}{g} . \ddot{x}_{(MAX)} \text{——} (1)$$

$$g = 386 \text{ in./sec.}^2$$

Equation (1) gives the minimum value of the spring pressure to prevent the roller jumping due to inertia.

The value of $\ddot{x}$ (max.) is very easily calculated in one case, viz., when the cam is so designed that the valve describes simple harmonic motion ; that is, when the graph of x and t is a sine curve. In this case it is convenient to measure x from the position of mid-lift positive outwards and negative inwards. The equation of motion is then :—

$$x = A.\sin pt \text{——}(2)$$

where A = Half the maximum lift in inches,

and p is a constant such that $pT = 2\pi$,

where T is the whole period of opening in seconds. If the exhaust valve is open for 240 crank-shaft degrees, then :—

$$T = \frac{60}{n} \times \frac{240}{360}, \text{ and } p = \frac{2\pi.n \times 360}{60 \times 240} = 0{\cdot}157n \text{——}(3)$$

" n " being the number of revolutions of the engine per minute.

From (2) $\ddot{x} = -Ap^2.\sin pt.$

And $\ddot{x}_{(MAX)} = -Ap^2.$

Substituting in (1) $X = \frac{W}{g}.A.p^2 \text{——}(4)$

Example : W = 10 lb.

A = Half lift = 0·375″.

n = 400 R.P.M.

From (3) $p = 0{\cdot}157 \times 400 = 62{\cdot}9.$

From (4) $X = \frac{10}{386} \times 0{\cdot}375 \times 62{\cdot}9^2 = 38{\cdot}5$ lb.

The valve would therefore require to be provided with a spring capable of exerting a force of 48·5 lb. to deal with inertia and dead weight only, apart altogether from gas pressure and friction.

The use of the harmonic cam, to which the above figures apply, has not become very general in Diesel Engine practice, the tangent cam shown in Fig. 228, Chapter XIII, being more commonly employed. Typical velocity and acceleration curves for a tangent cam are shown in Fig. 198.

Effect of Levers and Push Rods, etc.—The simple case considered above of a cam operating directly on the end of the valve is seldom realised in practice, and a somewhat more general case, illustrated diagrammatically in Fig. 199, will be considered. We have now several different members, all participating in the motion and acquiring momentum which

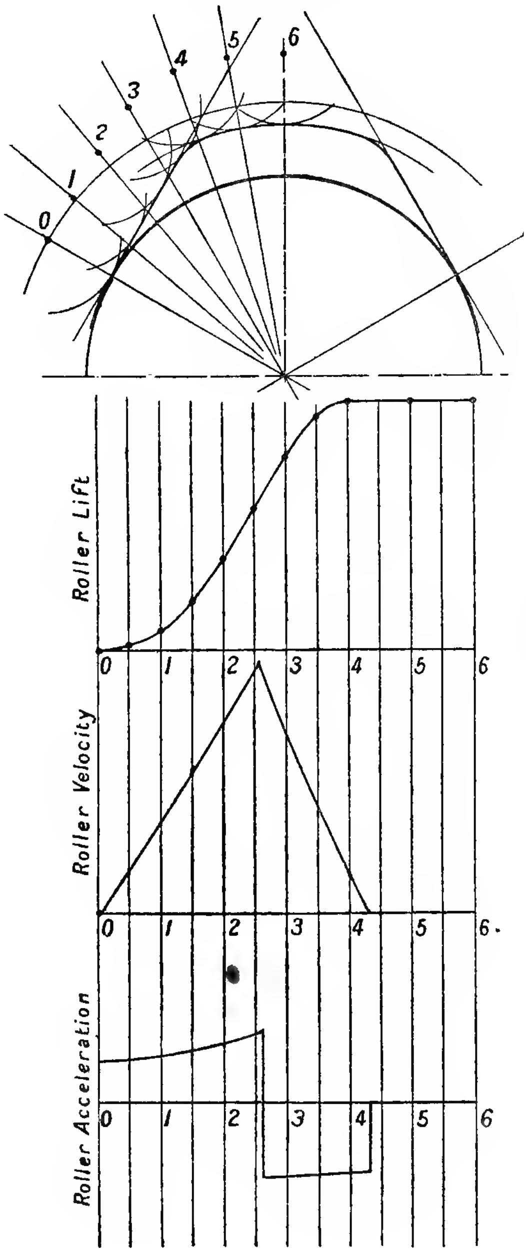

0
1
2
3
4
5
6
Roller Lift
0 1 2 3 4 5 6
Roller Velocity
0 1 2 3 4 5 6
Roller Acceleration
0 1 2 3 4 5 6

FIG. 198.

must be overcome by the spring. The problem is a simple case of the general theory of a system of one degree of freedom simplified by treating the "coefficient of inertia" as a constant instead of a function of x. Consider any particle of the lever AB situate at a distance "r" from the fulcrum E and having

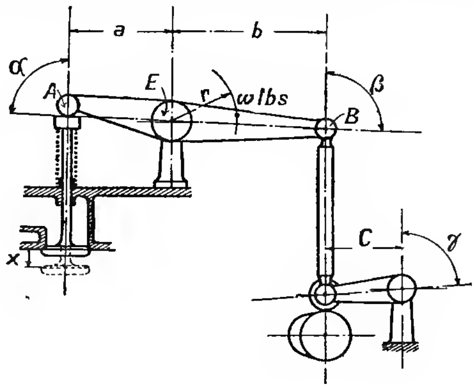


FIG. 199.

a weight "w" lb. If $\dot{s}$ denote the speed of this particle during any small displacement of the system, then :—

$$\dot{s}=\dot{x}.\frac{r}{a} \text{ and } \ddot{s}=\ddot{x}.\frac{r}{a}$$

The effective force acting on the particle is therefore equal to $\frac{w}{g}.\ddot{x}.\frac{r}{a}$ and the reaction X_1 at A due to all such particles of the lever is given by

$$X_1=\frac{\ddot{x}}{g\,a^2}\,\Sigma\, w.r^2=\frac{\ddot{x}}{g}.W_1\frac{k_1{}^2}{a^2}\text{——}(5)$$

Where W_1 is the weight of the lever and k_1 is its radius of gyration about E. The expression $\frac{W_1 k_1{}^2}{g.a^2}$ is the inertia coefficient of the lever with respect to the co-ordinate x, and may be denoted by A_1. The total reaction X at A, due to the inertia effects of the valve itself, all the levers, push-rods, etc., is the sum of all the reactions due to the individual members, and therefore,

$$X=\ddot{x}\,(A_0+A_1+A_2+A_3)\text{——}(6)$$

$$A_0 \text{ being}=\frac{W_0}{g} \text{ where } W_0=\text{weight of valve,}$$

and A_2 is the inertia coefficient of the push-rod, and A_3 is that of the link.

By similar reasoning to that given above for A_1 it is found that $A_2=\frac{W_2}{g}\left(\frac{b}{a}\right)^2$ where W_2=weight of push-rod.

and $A_3=\frac{W_3}{g}\left(\frac{k_3 b}{ac}\right)^2$, where W_3=weight of link, and k_3=its radius of gyration about its axis.

It will be seen at once that equation (6) is similar to (1), with inertia coefficient substituted for mass. The assumption made is that the angles α β γ do not deviate far from 90 degrees. For ordinary practical purposes a deviation of 10 or 15 degrees on either side involves a negligible error.

Example: $W_0=6$ lb. $a=10''$. $k_1=7''$
$W_1=15$ lb. $b=12''$. $k_3=5''$
$W_2=8$ lb. $c=7''$.
$W_3=5$ lb. $\ddot{x}=1600$ in./sec.^{2}

$$X=\ddot{x}(A_0+A_1+A_2+A_3)$$
$$=\frac{1600}{386}\left[6+15\left(\frac{7}{10}\right)^2+8\left(\frac{12}{10}\right)^2+\left(\frac{12\times5}{10\times7}\right)^2\right]$$
$$=118 \text{ lb.}$$

For slow running engines the inertia does not usually amount to more than two or three lb. per sq. in. of valve area. In most cases the spring may be based on the inertia load plus about 6 lb. per sq. in. of valve area, to deal with the other effects which enter into the question.

Strength and Deflection of Springs.—The usual formulæ for the safe load and the deflection of springs made of steel wire of circular section are given below for handy reference.

$P=0\cdot2f\cdot\frac{d^3}{r}$ Where
- P=Safe load (maximum) in lb.
- f=Safe stress, usually about 60,000 lb. per sq. in.
- d=Diameter of wire in inches.
- r=Mean radius of coils in inches.

And

$\delta=\frac{64.n.r^3}{d^4}\times\frac{P}{G}$ Where
- δ=Deflection in inches.
- n=Number of turns (free).
- G=Modulus of rigidity, usually taken to be about 12,000,000 lb. per sq. in.

Exhaust Piping.—A common arrangement of unjacketed cast-iron exhaust piping is shown in Fig. 200. The flexibility of the system renders any special provision for expansion unnecessary. The piping itself being out of reach need not be lagged, and may be as light as casting considerations will allow.

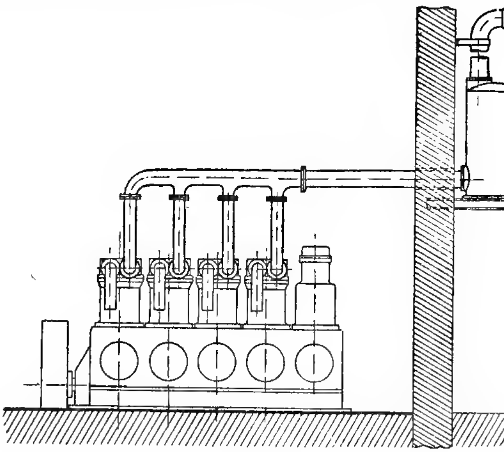


Fig. 200.

The connecting pieces between the various covers and the common discharge pipe may be made equal in bore to the diameter of the exhaust valve. The bore of the collector pipe joining the silencer may be proportioned with reference to the nominal velocity of the exhaust gases as follows :—

Let V_p = Piston speed in feet per second.
V_x = Nominal speed of exhaust in feet per second.
B = Bore of cylinder in inches.
d = Bore of exhaust pipe in inches.
n = Number of cylinders.

Then

$$V_x = V_p \left(\frac{B}{d}\right)^2 \times \frac{n}{4}$$

In using this formula "n" should be put equal to 4 in all cases where the number of cylinders is equal to or less than 4. The reason for this is that the gases discharge intermittently, and an engine of one, two or three cylinders requires approximately the same size of pipe as a four-cylinder engine of the same size and speed. The value of V_x varies from about 70 in

small engines to 110 in large. The pipe leading from the silencer to the atmosphere may be made about 25% larger in the bore. A somewhat neater arrangement, involving an intermediate collector under the floor, is shown in Fig. 201. The individual exhaust pipes must now be water-cooled, but there is no objection to short, uncooled sections in way of the flanges of sufficient length to accommodate the bolts. If the pipes are cast, the thickness of the outer walls need not exceed about $\frac{1}{4}''$ to $\frac{3}{8}''$ with good foundry work. The jackets may also be of welded steel tubes or sheets.

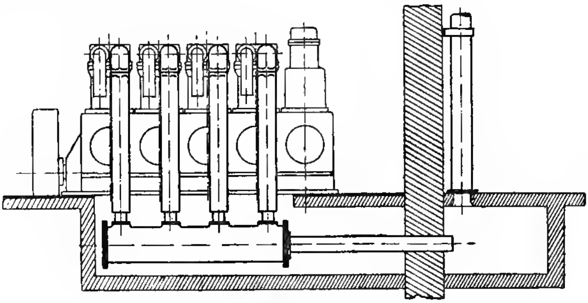


Fig. 201.

In marine installations the exhaust is sometimes used to furnish a supply of hot water for heating and other purposes, and this may be achieved by providing the exhaust collector with nests or coils of tubes through which water is circulated. As a rule the exhaust from a four stroke marine engine is not sufficiently hot to necessitate the provision of a water-cooled silencer apart from the arrangements which have just been mentioned.

Silencers.—As a rule four stroke Land Diesel Engines are supplied with a cast-iron silencer, having a capacity of about six times the volume of one of the working cylinders, and a common type is shown in Fig. 202.

The result is not always satisfactory, and better results are obtained by using a large underground brick or concrete chamber having twenty or thirty times the volume of one cylinder. Wrought iron silencers, unless water-jacketed or buried underground, usually give out a ringing noise, unless the gases on entry are made to diffuse through a trumpet arrangement similar to that described above under suction pipes.

Two Stroke Engines. Scavengers.—If the efficiency of the scavenging process could be definitely ascertained in every case, it would be a simple matter to calculate a suitable capacity for the scavenge pump. In those engines of which the results on trial suggest that this process is nearly perfect the scavenge pump appears to have a stroke volume capacity of about 1·4 times the aggregate stroke volume capacities of the cylinders which it feeds. It hardly appears safe, however, to assume that this allowance will necessarily be sufficient in any proposed design, and some constructors have adopted a higher figure, 1·8 to 2, in their first attempts at two stroke design. The scavenge air pressure is a factor which decides itself when once the details of port and valve openings have been fixed, and check calculations must be made on some such lines as those indicated in Chapter III, to make sure that reasonable limits of pressure will not be exceeded. The pressures obtained in practice vary from about 3 to 7 lb. per sq. in.

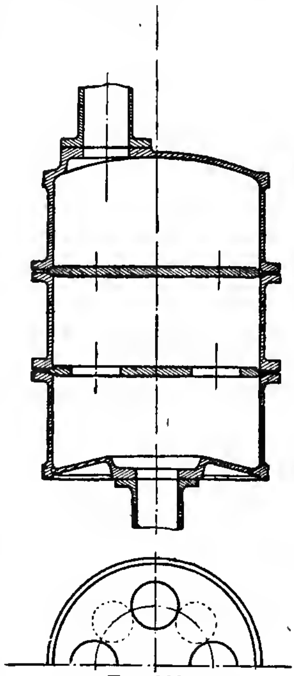


Fig. 202.

The work done by the scavenger is all lost work and must therefore be kept at a low figure. On the other hand, the super-pressure above that of the atmosphere with which a controlled scavenge engine starts the compression stroke, is a valuable factor in increasing the power of a given-sized cylinder. If simplicity is the main consideration and uncontrolled port scavenge is adopted, it becomes necessary to be content with a very moderate M.I.P., as the super-pressure obtainable at the beginning of compression appears to be very limited with the arrangements hitherto adopted on such engines.

Construction of Scavengers.—It is not necessary to deal exhaustively with the details of scavenge pumps as these differ little from the corresponding parts of L.P. steam engines. The scavenger, or scavengers, are preferably driven off cranks provided for the purpose on the main shaft, as in Fig. 203. In

marine designs the link-drive shown in Fig. 204 has also been used, but as usually carried out is open to the following objections :—

(1) The arrangement does not lend itself readily to the closed engine type of framework which appears to have every advantage for Diesel Engine work.

(2) The side levers involve cantilever connections, which lack rigidity.

(3) The heavy reversals of thrust are liable to cause knocking and vibration, owing to small bearing surfaces and poor lubrication.

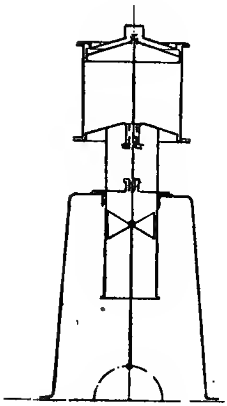


FIG. 203.

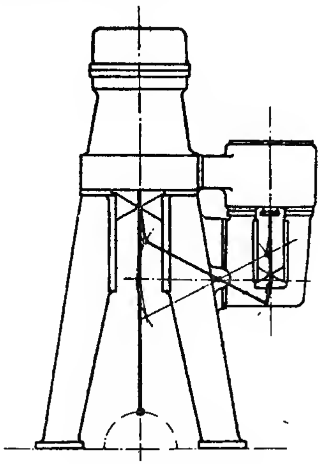


FIG. 204.

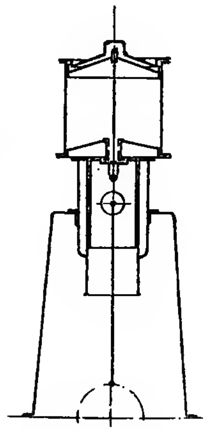


FIG. 205.

A scheme suitable for land work but a trifle inaccessible for marine purposes consists of a tandem arrangement of scavenger and low pressure stage of blast air compressor, as in Fig. 205.

Valve Gear.—The earlier Diesel scavengers were fitted with piston valves single or double ported, and this type of gear is still retained in some designs. For reversing such valves the Stevenson link motion appears to be the best solution on hand at present. In other designs automatic disc or plate valves are being increasingly used, with a gain in simplicity and the advantage in marine work that no reversing gear is necessary. The liability of the valves to failure by fatigue seems to be their chief drawback.

Scavenge Air Receivers.—From the scavenge pump the air

passes to a receiver in communication with the cylinder covers or scavenge air belts of the working cylinders. With the usual arrangement, where one, or at most two, double-acting scavenge pumps are used to supply a number of working cylinders, say three to eight, it is advisable to make the capacity of the receiver large compared with the stroke volume of one scavenger, in order that the pressure in the receiver may remain sensibly constant, and a suitable capacity may usually be secured by making the diameter of the receiver about 1 to 1·3 times the cylinder bore. In any proposed case it is a simple matter to construct a diagram showing on a time base the rate at which air is being passed to the receiver by the scavenger and carried away from it by the working cylinders. The resultant effect in creating fluctuations in a receiver of any proposed capacity is then easily calculated. The effect of too small a receiver capacity is to give those cylinders which begin compression at the instant of maximum scavenge pressure an advantage over those less favourably timed. The effect is readily studied in practice by means of light spring diagrams taken from the receiver and the working cylinders respectively. By way of example, if two cylinders start compression with absolute pressures of 21 and 20 lb. per sq. in. respectively, then the first will (other things being equal) have a maximum power capacity 5% greater than the second, or if they are worked at the same power the second cylinder will work with a mean absolute charge temperature 5% greater than the first, which is no small evil. In some two stroke designs one double-acting scavenger is provided for each pair of working cylinders, and if the deliveries are correctly timed with respect to the scavenge periods the receiver capacity does not require to be very large. Scavenge receivers are usually made of riveted or welded sheet steel, having a thickness of about one per cent of the diameter. A disastrous explosion at Nuremberg in 1912, traceable to the ignition of lubricating oil vapour in the scavenge receiver of a large two stroke engine, points to the desirability of providing drain cocks and a relief valve or safety diaphragm of large area. The area of the trunk communicating with each cylinder should be well in excess (usually about double) of the maximum aggregate area of the valves or ports which it supplies.

Scavenge Valves.—In different designs embodying the principle of scavenging through the cover one, two, three and four valves have been employed. In particular, if two valves

are used they may be identical with the air and exhaust valves used in four stroke engines of the same size, with the piston speeds at present customary. With any other number of valves the area may be made equivalent. In view of the

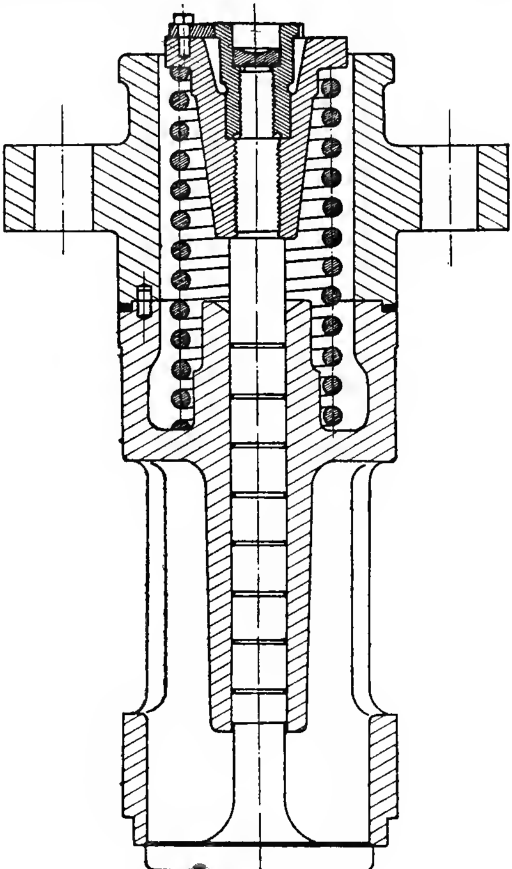


Fig. 206.

relatively light duty which devolves upon them, scavenge valves are usually made of somewhat simpler construction than exhaust valves, as shown in Fig. 206. To prevent loss of scavenge air past the spindle the latter is usually made a good fit.

With controlled port scavenge, double-beat valves, piston

valves and Corliss type valves have all been used in different designs, as shown diagrammatically in Fig. 207. The use of the valve is exclusively to regulate the instant at which air is admitted to the cylinder, the point of cut-off being determined by the piston covering the ports on the up stroke.

It therefore follows that so long as the valve is full open at the instant when the ports are covered, the point at which the valve seats again, may be determined arbitrarily by other considerations affecting the valve gear.

Exhaust System.—The chief evil to be guarded against in the design of the exhaust system of a two stroke engine is the interference of the exhaust rush of one cylinder with the scavenge process of another. An obvious but inconvenient way of avoiding this trouble is to provide separate exhaust

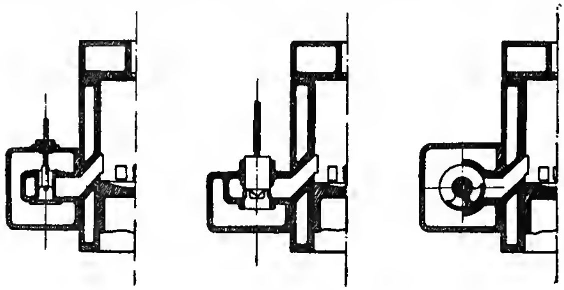


FIG. 207.

pipes and silencers for each cylinder. Practically the same effect can be achieved by providing common exhaust systems for pairs of cylinders whose cranks are at 180°. This, however, does not appear to be necessary, and satisfactory results with an exhaust system common to all cylinders are obtained by the use of arrangement similar to that shown in Fig. 201 for four stroke engines.

The essential point is that the pipes which connect each cylinder to the common collector should be of large diameter (about three-quarters of the cylinder bore), and that the collector itself should have a volume large in comparison with the stroke volume of one cylinder. It does not follow, however, that a free passage for the exhaust is a necessary condition for efficiency, as the latter has sometimes been improved by the insertion of a throttling diaphragm in the exhaust passage at the point where the pipe joins the cylinder.

Silencers.—The sudden release by the uncovering of ports of a pressure of 40 lb. per sq. in. and upwards produces a noise which in the absence of a silencer can be heard some miles off. A type of silencer which renders the exhaust inaudible a few yards away without imposing any back pressure, is shown in Fig. 208. This type of silencer is most effective when water-

Diameter of Silencer . . .	3·5 B
Length ,, ,, ...	5·5 B
Bore of Inlet Pipe	0·55 B

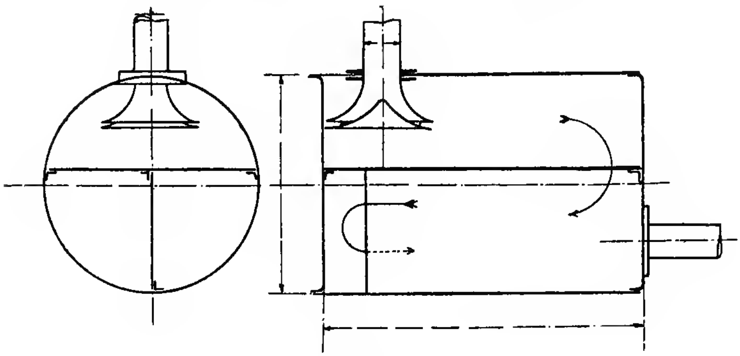


Fig. 208.

jacketed. Approximate main dimensions are given in terms of the bore of the engine cylinder on the assumption of a piston speed of about 800 feet per minute. For land purposes a large pit without special baffles would probably serve equally well.

Literature.—For information on the mechanics of cam-operated mechanism, see :—

Goodman, J., *Mechanics Applied to Engineering* (Longmans).

CHAPTER XII

COMPRESSED AIR SYSTEM

As mentioned in Chapter I, the injection of fuel by means of an air blast is one of the outstanding characteristics of the Diesel Engine, and it seems probable that its use in the early experimental engines was suggested by the compressed air apparatus used to start the engine, and which still appears to be the most practicable method of doing this. The utility of the air blast is by no means confined to its function of injecting the fuel; in fact the widespread use of mechanical means of injection in other types of oil engine clearly indicates that effective atomisation can be obtained otherwise. As Guldner has pointed out, the use of an air-blast probably secures a more efficient mixing of the cylinder contents than could be obtained in any other practicable manner. The advantages in efficiency which such mixing secures are easily appreciated on examination of the thermodynamic principles involved. With good mixing the combustion proceeds rapidly, and reduces after-burning to a minimum; and, further, the whole charge tends to remain homogeneous as to temperature, a necessary condition for maximum efficiency. In practice there are two aspects from which the efficient utilisation of heat should be viewed, viz. :—

(1) Economy in fuel consumption.
(2) The effect of efficient combustion in keeping the mean cycle temperature to a minimum.

The last consideration is a vital one from the point of view of reliability and durability, and experience abundantly proves that a relatively small increase of the cycle temperature, due to overloading, leakage past valves, loss of volumetric efficiency or other causes is sufficient to convert an otherwise reliable machine into a source of continual trouble.

The blast air has a pressure which varies from about 900 to 1000 lb. /sq. in. at full load, to about 600 lb. /sq. in. at no load, and in land engines is usually supplied by a compressor forming an integral part of the engine. In marine installations the compressors are sometimes driven by separate auxiliary engines. The arrangement adopted by one maker is to drive the lower stages of the compressors by auxiliary engines, the last stage being performed by a high pressure plunger driven by the main engine. From the compressor the air passes, via coolers, to a blast reservoir or bottle of sufficient capacity to absorb fluctuations of pressure, and fitted with suitable distributing valves, one of which communicates with the fuel injection valves and another enables surplus air to be passed to the storage reservoirs provided for starting purposes. Cam-operated valves in the covers of one or more cylinders enable the stored air to be used, to give the engine the initial impetus which is necessary before firing can begin. With land engines the starting bottles are generally charged to a pressure of about 900 lb. per sq. in., and with the fly-wheels commonly used it is not necessary to provide starting valves for more than one cylinder out of three. With marine engines, storage pressures of about 300 lb. per sq. in are more common, on account of the difficulty of making high pressure reservoirs of large size, and in order to secure prompt starting from any position starting valves are fitted to every cylinder.

The air system also includes certain servo-motors or air engines, frequently used to perform operations of reversing the valve gear. The various organs will now be considered in more detail.

Air Compressors.—Four stroke land engines of the slow speed type as at present constructed require compressors having a capacity of about 15 cubic feet per B.H.P. per hour, which assuming a volumetric efficiency of 80% corresponds to a stroke volume capacity of about 19 cubic feet per hour. High speed engines appear as a rule to require about 25% more than this allowance. The above method of basing the compressor capacity on the B.H.P. is not a very satisfactory one, as different makers have different views as to power rating. A better plan is to express the L.P. stroke volume as a percentage of the aggregate cylinder volume, and the following figures are representative of average practice for land engines. In view of the demands made on the system when manœuvring,

marine engines are usually provided more liberally, to the extent of 50 to 100%.

Bore of working cylinders.	Ratio L.P. stroke vol. ÷ Stroke vol. of working cylinders.	
	Four Stroke Engines.	Two Stroke Engines.
10	0·08	0·16
15	0·07	0·14
20	0·05	0·09

Number of Stages.—For small slow-running compressors two stages are sufficient, but a 9-inch diameter of low pressure cylinder appears to be about the safe limit ; and even with this restriction it appears wise to abandon the principle of equal distribution of work between the stages. The small diameter of the H.P. cylinder affords little cooling surface for the dissipation of heat, and this consideration points to the advisability of arranging for the greater part of the work to be done in the L.P. stage, a conclusion which has been anticipated by experience.

Three-stage compressors are being increasingly used, even the smaller sizes, four stages being required in the very largest installations only. With three or four stages the principle of equal division of work is open to less objection, owing to the smaller ratio of compression in each stage.

Compressor Drives.—Almost every conceivable type of drive has been adopted at one time or another, and only the commonest are mentioned below :—

(1) Tandem two or three stage compressor driven off the crank-shaft. This arrangement appears to have the balance of advantages for most purposes.

(2) Tandem two-stage compressor driven by links and levers from each connecting rod or crosshead. This arrangement is expensive, but has the advantage of distributing the work amongst a number of small compressors, which are subject to less heat trouble than one compressor of the same capacity. The suction pressure of the H.P. stage is usually sufficient to prevent reversal of thrust due to inertia, and consequently sweet running is secured.

(3) Similar to (2), but stages separate. This arrangement is bad, as the cooling surface is less than in case (2), and the L.P. gear is subject to reversal of thrust due to inertia.

(4) Twin tandem cylinders driven by links off the crank-shaft. This arrangement gives good results, as the load is divided between two units and the pressure on the crank-pin is reduced by the leverage of the link-work.

Constructive Details.—The construction of air compressors being a specialised branch of mechanical engineering, it is not proposed to give here more than a very brief reference to the subject. The cylinders of tandem two and three stage machines are frequently cast in one piece, including the water-jacket. The relatively low temperatures obtaining justify this procedure, provided sound castings can be obtained with reasonable regularity. The foundry work may be simplified in the case of two-stage compressors by the following division of material :—

L.P. Cylinder and Jacket—one casting.
L.P. Cover and H.P. Jacket—one casting.
H.P. Liner and H.P. Cover—separate castings.

One advantage of this scheme is the possibility of renewing the H.P. liner when worn. The latter is peculiarly subject to rapid wear on account of the high pressure behind the rings.

Similar arrangements are of course possible with three-stage machines. The possibility of increasing the cooling surface of the H.P. liner by means of ribs does not appear to have received very much attention. The trunk pistons serve as admirable crossheads, being almost entirely free from the heat troubles to which the pistons of internal combustion engines are subject. For the L.P. and intermediate stages Ramsbottom rings are usually fitted. In very large machines the latter can also be fitted to the H.P. plunger. Small H.P. plungers are usually fitted with some arrangement similar to that shown in Fig. 209. The only thing which need be said about the connecting rods is that on account of the thrust being always in one direction, special care is required in the details of lubrication.

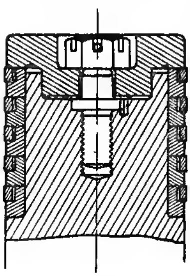


FIG. 209.

The design of valves has an important bearing on the success

or failure of a compressor. The chief evils to be avoided are :—

(1) Sticking of the valves off their seats, due to deposits of carbonised oil.

(2) Damage to valves or valve seats, due to hammering.

The first is influenced more by the efficiency of the cooling arrangements and the compression ratio than with the design of the valves themselves. For obvious reasons the H.P. valves are most subject to this trouble.

The second trouble is usually due to the valves being too heavy, having too much lift, or the failure to provide adequate cushioning, and in successful designs is avoided by one or more of the following means :—

(1) Making the valves in the form of very light plates or discs with a very small lift.

(2) Providing a large number of very small valves in place of one or two large ones.

(3) Where large valves of considerable weight are used, arranging for some sort of dash-pot action.

All the valves should be easy of access and removal. Experiments with existing types of compressor seem to indicate that makers are inclined to base their valve dimensions on an air speed very much lower than is necessary. One or two per cent loss of efficiency is of small importance, if such a sacrifice enables the size of the valves to be reduced.

In some designs the intercoolers are separate from the compressor cylinder, and in others take the form of pipe-coils arranged round the compressor cylinders inside a removable water-jacket. Vibration of the coils should be prevented by adequate clamps and stays, and no sharp bends are allowable, on account of a scouring action (presumably due to turbulent flow) which in acute cases may cause fracture of the pipe in a short time. L.P. intercoolers are sometimes made similar to tubular condensers, and in other designs take the form of a cast-iron vessel provided with internal helical baffles which give rise to turbulent flow and increase greatly the efficiency of the cooling surface. It is desirable in all cases to fit a final cooler, to reduce the temperature of the fully compressed air before entering the blast receiver. Each receiver should be fitted with safety valve and drain. One or two isolated cases of explosion, traceable to accumulation of lubricating oil in the intercooler system, emphasise the necessity for these fittings.

Some makers fit special "purge-pots" in communication with each receiver for the collection and discharge of condensed water and oil.

Calculations for Compressors.—In calculating the L.P. stroke volume required to furnish a given free air capacity, allowance must be made for the volumetric efficiency, which depends mainly on the clearance space and the delivery pressure. For example, suppose the clearance to be 3% of the stroke volume and the receiver pressure to be 150 lb. per sq. in. On the suction stroke, the suction valve will not begin to lift until the air left in the clearance space has expanded down to atmospheric pressure. If this clearance air expands according to the law :—

$$P.V.^{1\cdot 2}=\text{constant},$$

then its expanded volume expressed as a percentage of the stroke volume will be :—

$$3\times\left(\frac{164\cdot 7}{14\cdot 7}\right)^{\frac{1}{1\cdot 2}}=22\cdot 5\%$$

Subtracting its original volume, viz., 3%, the amount by which the effective stroke is shortened is 19·5%, and the volumetric efficiency is 80·5%. A further deduction should strictly be made for the fact that at the beginning of the compression stroke the cylinder contents are in general at a pressure slightly less than atmospheric. One or two per cent will usually cover this contingency. This example is sufficient to show the importance of reducing the clearance volume of the L.P. cylinder to a minimum. The efficiencies of the L.P. or H.P. cylinders may be found similarly, but only influence the volumetric efficiency of the compressor as a whole indirectly by raising the receiver pressure above the value it would have if there were no clearance. It will be evident on reflection that leakage past the H.P. delivery valves will also raise the receiver pressures, and for this reason it is desirable to fit pressure gauges to all receivers so that the condition of the valves may be inferred from the gauge readings.

Assuming perfect intercooling and equal volumetric efficiency in all the stages, the pressure of the atmosphere and the receiver pressures (absolute) will be in inverse ratio to the stroke volumes of the cylinders, and equal division of work between the stages will be secured by the proportions given below :—

	Atmospheric Pressure.	Intermediate Pressure.	High Pressure.
Two Stages.	1 at. = 14·7 lb./in.^{2}	8 at. = 118 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume = 8 H.P. volume.

	Atmospheric Pressure.	1st Intermediate Pressure.	2nd Intermediate Pressure.	High Pressure.
Three Stages.	1 at. = 14·7 lb./in.^{2}	4 at. = 58·8 lb./in.^{2}	16 at. = 235 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume : I.P. volume : H.P. volume = 16 : 4 : 1.

In practice better results are obtained with two-stage machines by the following proportions :—

Atmospheric Pressure.	Intermediate Pressure.	High Pressure.
1 at. = 14·7 lb./in.^{2}	12 at. = 177 lb./in.^{2}	64 at. = 940 lb./in.^{2}

L.P. volume = 12 H.P. volume.

Assuming that the L.P. stroke volume has been determined by some such considerations as the above, the actual cylinder dimensions are found by selecting suitable values for the piston speed and the L.P. stroke to bore ratio. The piston speeds commonly used lie between about 300 to 600 feet per minute, the lower speeds being usually associated with small machines. With a little care it is possible, by making small variations in the strokes, to design a series of four or five compressors suitable for engines covering a wide range of powers. Such a scheme involves sacrifices in some cases which, however, would appear to be quite outweighed by the advantages of standardisation. The ratio of stroke to bore of the L.P. cylinder will generally lie between about 0·7 and 1·7. The valve areas for each stage are based on some figure for the mean velocity obtained, in the case of suction valve, by multiplying the mean piston speed by the ratio of piston to valve area, and in the case of delivery valves the mean piston speed during the delivery period by the same ratio. Certain continental authorities recommend speeds not exceeding 80 and 115 feet per second for the suction and delivery respectively, but it appears that these figures may be doubled or even trebled with impunity, and sometimes to advantage.

The calculations of the strength of the various parts are straightforward, involving no special principles, and are therefore passed over. The cooling surface to be provided for intercooling is a very important matter, and the following figures from a successful design may be useful as a basis of comparison in the absence of first-hand experimental data.

THREE-STAGE COMPRESSOR. Free air capacity, 130 ft.^{3}/min.

Cooling surface, copper-pipe coils.
- L.P. 8·7 ft.^{2}
- L.P. 3·6 ft.^{2}
- H.P. 3·6 ft.^{2}

The subject of heat transmission being a very important one in connection with internal combustion engines, a brief reference to the usual theory is inserted below.

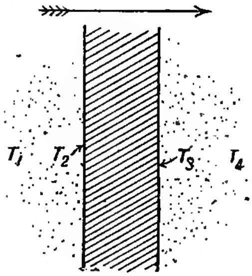


FIG. 210.

Transmission of Heat through Plates.—Referring to Fig. 210, the direction of heat flow is indicated by the arrow and the symbols t_1, t_2, t_3, t_4 denote the temperatures of the hot fluid, the hot side of the plate, the cold side of the plate, and the cold fluid respectively. The total heat drop consists of three stages :—

(1) An apparently sudden drop from the hot fluid to the plate.
(2) A steady gradient across the thickness of the plate.
(3) An apparently sudden drop from the plate to the cold fluid.

The assumption is that the rate of heat flow is dependent only on the temperature drop, the thickness of the plate, and the particular fluids employed. On this assumption, if Q is the amount of heat transmitted per hour per unit of area, then :—

$$Q=\alpha_1(t_1-t_2)=\frac{\lambda}{d}(t_2-t_3)=\alpha_2(t_3-t_4) \text{——} (1)$$

from which

$$Q=\frac{(t_1-t_4)}{\frac{1}{\alpha_1}+\frac{1}{\alpha_2}+\frac{d}{\lambda}} \text{——} (2)$$

Where d = thickness of plate and α_1, α_2, and λ are constants.

According to Hütte, for air at atmospheric pressure and small velocities α_1 = about $0{\cdot}4+1{\cdot}1\sqrt{v}$.

Where v = velocity in ft./sec., and for water (not boiling and without turbulence)

$$\alpha_2 = \text{about } 100 \text{ B.T.U./ft.}^2 \text{ deg. F.}$$

Values of λ are given below for various metals :—

Iron . . .	460	B.T.U. /ft.^{2} deg. F. per inch of thickness.
Mild steel . .	320	
Copper . . .	2100	
Brass . . .	740	

The values of λ are well determined, but unfortunately the term involving this constant is the least important of the three as the bulk of the heat drop occurs at the surfaces of the plates. The values of a_1 and a_2 must be used with caution as the small amount of published data indicates that these constants are subject to enormous variation under different circumstances. In particular, the value of a_2 is greatly increased by the eddying motion produced by the introduction of spiral baffles in condenser tubes and the like, and is increased about tenfold if the water boils. The value of a_1 for air increases greatly with rise of temperature and pressure.

Apart from these uncertainties, formula (2) suggests the following corollaries :—

(1) The rate of heat transmission is but little influenced by the thickness of the plate in most practical cases, such as cylinder covers, etc.

(2) The temperature drop across the plate, and consequently the temperature stress, is proportional to the rate of heat transmission and to the thickness of the plate ; this suggests one reason why it is advisable to work large engines at a smaller mean cycle temperature than small ones.

For further information on the subject of heat transmission, the reader is referred to the sources of information mentioned in the footnote.[1]

Air Reservoirs.—The usual arrangement of air reservoirs for land engines is shown in Fig. 211. This scheme was devised in the very early days of the development of the Diesel Engine and no substantial improvement has been made on it in recent years. Two starting and one blast-air bottles are provided, all designed for a working pressure of about 1000 lb. per sq. in. One of the starting bottles serves as a reserve, in case of a

[1] "High-speed Internal Combustion Engines," Judge. "Heat Transmission," Report by Prof. Dalby to the Inst. Mech. Engs., 1909. "Notes on Recent Researches," paper by Prof. Petavel: Manchester Assoc. of Engineers, Oct., 1915. "The Laws of Heat Transmission," Lecture by Prof. Nicholson: Junior Inst., Jan., 1909.

failure to start the engine, due to any derangement. In the event of such failure, every care is taken to make certain that the engine is in perfect order before using the reserve bottle, and it seldom happens in practice that the bottles require to be replenished from outside sources of supply. The connections between the bottles, the air compressor and the engine should

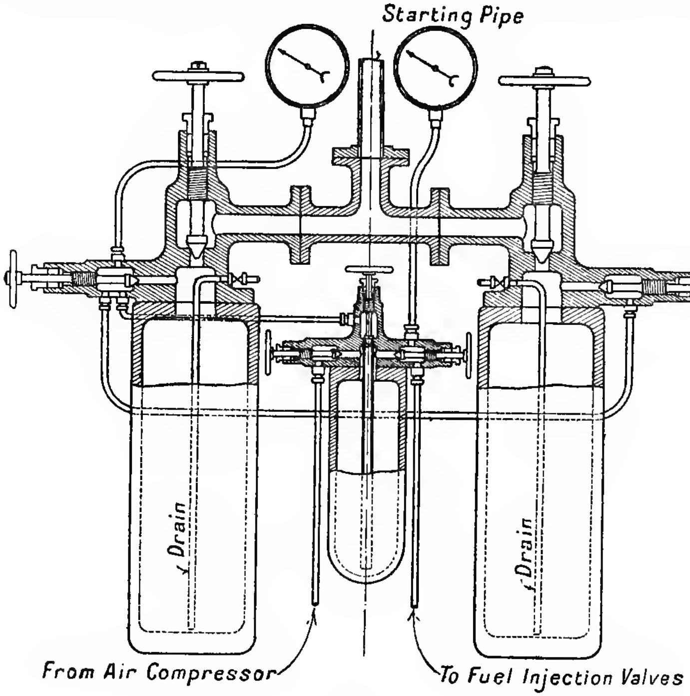


FIG. 211.

be quite clear from the diagram. Only one or two points will be mentioned.

(1) Before starting up, it is possible to ascertain the pressure in each of the three bottles by opening up the appropriate valve on each bottle-head in rotation In each case the pressure is recorded on the left-hand gauge.

(2) The pressure in any pair or all three bottles may be equalised by opening up a pair or all three such valves.

(3) The right-hand gauge registers the blast pressure on the engine side. By throttling the blast control valve on

the blast bottle-head the injection pressure may be regulated below that of the bottle. This is done when replenishing the starting bottle on light load. It is thus possible to pump up the starting vessel to 1000 lb./in.^{2} whilst the blast pressure is only 600 lb./in.^{2}, as required for light running.

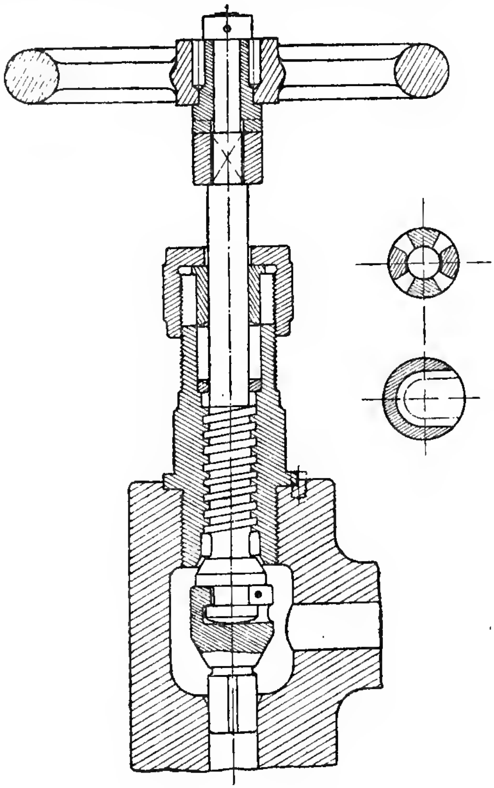


FIG. 212.

The bottle-heads containing the various valves are usually machined from a solid block of steel A detail of one of the valves is shown in Fig. 212.

The bottles themselves are of weldless steel, and a neck is frequently screwed on, as in Fig. 213. Some idea of the capacities of the bottles commonly provided may be gathered from the following table :—

TOTAL CAPACITY OF H P STARTING AIR BOTTLES
(four stroke engines)

Engines of about 9″ bore, having one to six cylinders—about fourteen times the stroke volume of one cylinder.

Engines of about 24″ bore, having one to six cylinders—about seven times the stroke volume of one cylinder.

Owing to the expensive machinery required to manufacture weldless reservoirs, only a certain limited number of standard sizes are available at reasonable prices, and in very large installations it is sometimes necessary to provide groups of four or more starting vessels. The usual working stress is about

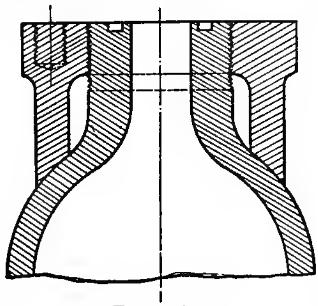


FIG. 213.

8000 lb./in.^{2}, and it is customary to specify a water-test pressure of double the working pressure.

Riveted Air Reservoirs.—For marine installations of high power it is usual to use a lower air pressure of about 300 lb./in.^{2} for starting purposes. The air reservoirs now require to have a very much larger cubic capacity, but the reduced pressure permits of the employment of riveted reservoirs. The construction of the latter need not be dealt with here, being comparable with that of the steam drums of modern water-tube boilers. Adequate drainage for condensed water and oil vapour, and also a manhole for inspection and cleaning, should be provided. These matters, as well as others dealing with the strength of the riveted joints, the quality of material to be used and the tests to be carried out on completion, form the subject-matter of regulations by the various insurance societies and the Board of Trade.

Blast Piping System.—From the blast bottle the injection

air passes to a main running along the back of the engine, where it is distributed by short lengths of pipe to the several fuel valves, as in Fig. 214. Where one fuel pump is provided for a number of cylinders the fuel distributors may be made to serve as distributing tee-pieces for the blast air. In marine

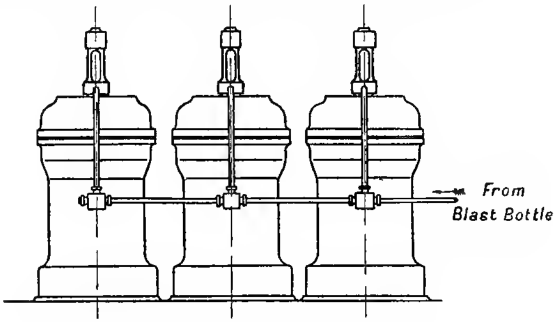


FIG. 214.

engines it is usual to provide a shut-down valve, as in Fig. 215, to each tee-piece, so that the supply to any individual cylinder may be cut off, to enable the fuel valves to be changed without stopping the engine.

In addition, it is sometimes necessary (see Chapter XIII) to provide a valve whereby the whole supply of blast air is automatically cut off when the manœuvring gear is put into the stop position. The bore of the blast air supply pipe to each cylinder need not exceed about 2% of the cylinder bore, but is usually greater than this in small engines, to avoid the multiplication of standard sizes of unions. The blast air main may be about 4% of the cylinder bore for any number of cylinders up to about six. The same type of union may be used as has already been illustrated in Fig. 156 in connection with the fuel system. Other types of union are in use, notably the Admiralty Cone Union, which is also very serviceable.

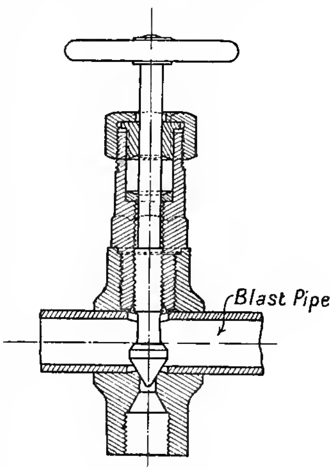


FIG. 215.

The Starting Air Pipe System.—With land engines it is quite common to provide one cylinder only with a starting valve when the number of working cylinders does not exceed

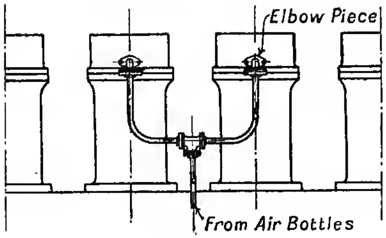


FIG. 216.

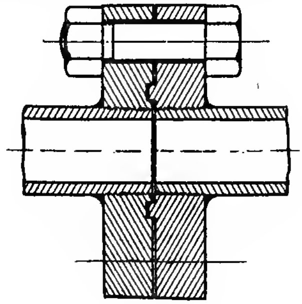


FIG. 217.

four. With six cylinders and upwards two and sometimes three units are provided with air-starting arrangements. A neat arrangement of the starting pipe is shown in Fig. 216 for a four-cylinder engine. In this case the design of the starting

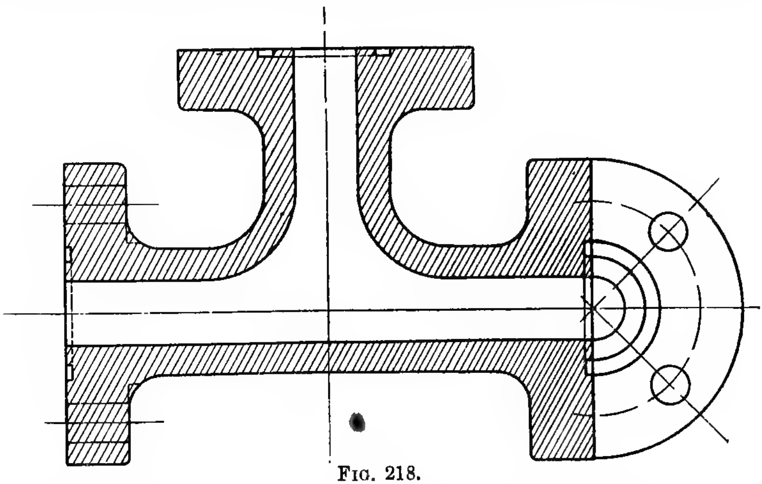


FIG. 218.

valve is such that air is admitted through a port cast in the side of the cylinder cover. Any arrangement of piping is to be avoided which renders difficult the removal of a cylinder cover, hence the provision of an elbow on the latter. In other designs this elbow is cast integrally with the cover itself.

With marine engines all cylinders are provided with starting valves, to which the air is led through a steel main pipe line running the whole length of the engine. Fig. 217 shows the type of pipe flange most commonly used, the material being steel. The tee-pieces for distribution to the several cylinders may be of cast iron or cast steel. If the former material is used, the design should be very substantial, as in Fig. 218. The pipe lines should be securely clipped to the framework of the engine, otherwise there is liable to be severe vibration, due to the surging of pressure within the pipe.

In large slow speed engines the diameter of the starting pipe may be about 0·07 to 0·1 of the cylinder diameter. In small high speed engines it is advisable to give the main distributing pipe a diameter of about 0·15 to 0·17 of the bore, in order to secure rapid acceleration.

Starting Valves.—These are usually located in the cylinder cover and operated by cams and levers, in the same way as the other valves. An arrangement less frequently used consists of a centralised air distributing box of rotary or other type remote from the cylinder covers but connected to them by distributing pipes. Loss of compression is obviated by the provision of non-return valves in the cylinder cover. The centralised distributing box may consist of a sleeve rotating in a casing in such a manner that a slot in the sleeve admits air successively to a number of ports communicating with the several cylinders. In other arrangements a set of cam-operated mushroom valves is used. These schemes have not become common practice and will not be discussed here in further detail.

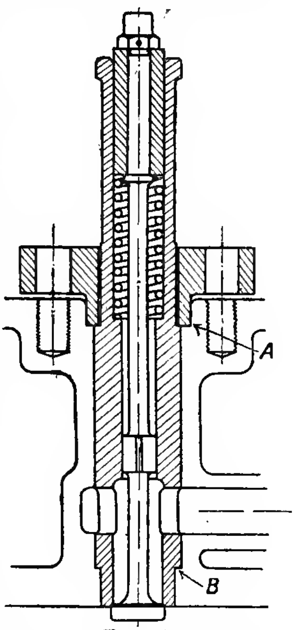


Fig. 219.

A common type of starting valve is shown in Fig. 219. No provision has been made here to prevent leakage past the spindle, and if the latter is a good ground fit in the casing the leakage should not be serious in amount. In large sizes of valve additional tightness may be secured to advantage by

the provision of a number of small Ramsbottom rings. It is usual to make the diameter of the piston part of the spindle the same as the smaller diameter of the valve head. The minimum spring compression should be equivalent to the maximum starting air pressure acting on an even area equal to that of the valve seat. The valve casing should be substantially proportioned, to prevent distortion and consequent leakage at the seat or binding of the spindle. It will be noticed that with this design of valve, joints have to be made at A and B simultaneously. There is no practical difficulty about this. Joint A is usually made with a copper or white-metal ring. A slight modification is sometimes made by the introduction of two cone joints, as in Fig. 220. This also works well. Fig. 221 shows a type of starting valve in which the air is led to the top of the valve casing, instead of being introduced through a port in the cylinder cover.

A useful type of starting valve, devised by the Burmeister & Wain Company for Diesel Marine Engines, is illustrated diagrammatically in Fig. 222. With this design the valve becomes inoperative when the air pressure is removed and resumes working as soon as the pressure is restored. This results in a great simplification of the manœuvring gear (see Chapter XIII) by the elimination of mechanism which in some other designs is provided for the purpose of throwing the starting valves out of gear when the fuel is turned on. The desired result is achieved by attaching to the upper end of the valve spindle an air cylinder and piston, kept in constant communication with the air supply by means of holes through the spindle. In the absence of air pressure, the spring A is sufficiently strong to keep the piston B at the bottom of the cylinder C, thus removing the roller from the range of operation of the cam D. When pressure air is turned on the piston is forced to the top of the cylinder, and the valve remains operative so long as the force required to open the valve is less than the difference between the pressure load and the spring load on the piston B.

Diameter of Starting Valves.—On theoretical grounds, the necessary diameter of starting valves would appear to depend on the pressure of the air supply, amongst other things. It so happens, however, that in those cases where a low pressure air system is the most convenient (viz. in large marine installations) the multiplicity of cylinders to which starting air is

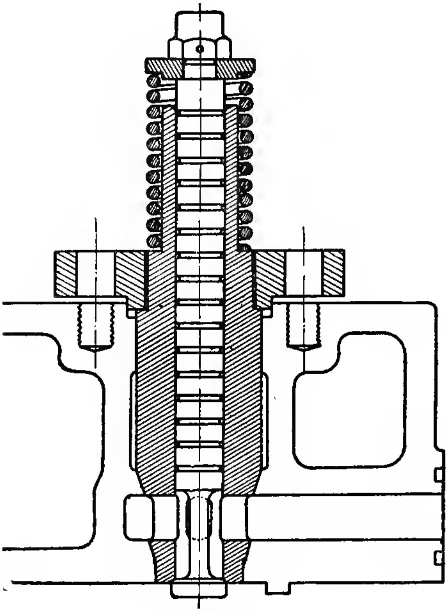


FIG. 220.

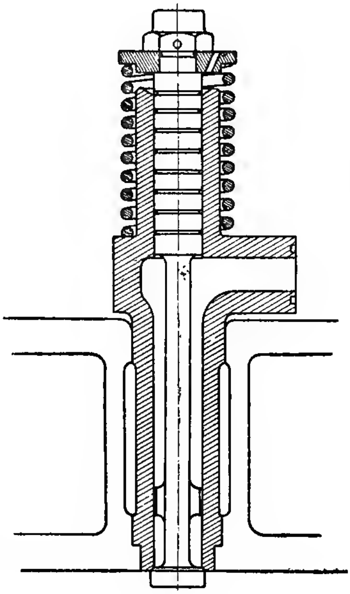


FIG. 221.

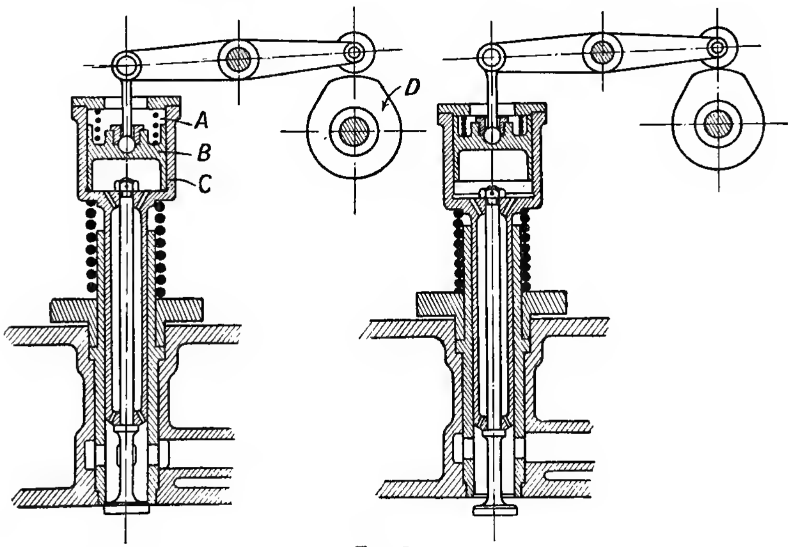


FIG. 222.

supplied affords adequate starting torque with a relatively low mean starting pressure in each cylinder. The result is that roughly the same diameters of starting valve are used in either case, i.e. whether a low or high pressure starting system be adopted, typical figures being from about 0·1 of the bore in the case of large engines to 0·13 in small engines.

With slow speed land engines it is very desirable to obtain a "fat" starting card, to overcome the inertia of the heavy flywheels which are usually necessary. An engine in good working order should start firing in the first or second revolution on starting up cold. It seems probable that in the event of low pressure air being used for such engines, it might be necessary to fit starting valves to all the cylinders or to make the diameter of the latter larger than is customary with the high pressure starting air systems at present in use.

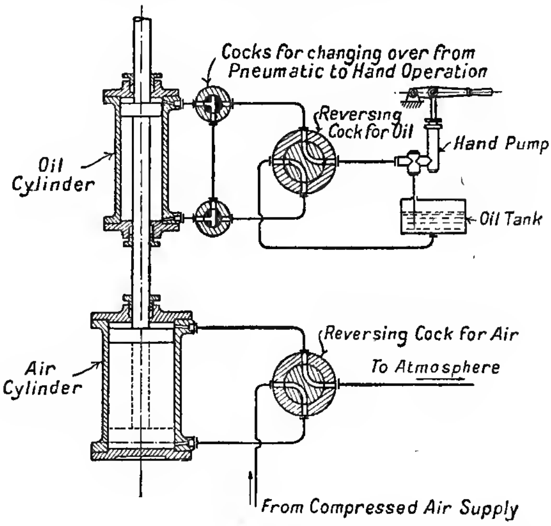


FIG. 223.

Air Motors.—In large marine engines the work required to effect reversal of the valve mechanism when going from ahead to astern, or vice versa, is generally too great to be done with sufficient rapidity by a hand gear, except in case of a breakdown of the air motor which is usually provided for the purpose. In different designs the air motor takes various forms, of which some are mentioned below :—

(1) A small reciprocating engine (double acting), with two cylinders and cranks arranged at right angles. The arrangement is almost exactly similar to the small auxiliary steam engines used on steamships for the reversing gear or the steering gear. Low pressure air is used, and the air motor is geared down by worm and worm-wheel, so that it makes a considerable number of revolutions for one movement of the reversing gear.

(2) A single cylinder, with piston and rod, the reversing motion being performed in one stroke. This arrangement is suitable for high or low pressure air, and in either case the piston-rod must be extended into an oil dashpot cylinder to reduce shock. If the two ends of

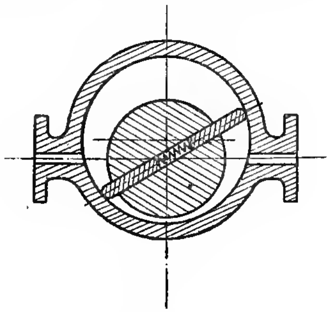


FIG. 224.

the oil cylinder be connected to a hand pump, the latter may be used for reversing, in the event of a failure of the air cylinder. This scheme is illustrated diagrammatically in Fig. 223.

The reciprocating motion of the piston-rod may be converted into rotary motion (one complete revolution or more) by a rack and pinion.

(3) A rotary engine of the type which is frequently used as a pump in connection with machine tools, motor-cars and other small machines, and which is shown diagrammatically in Fig. 224. This type of motor is only suitable for low pressures and is arranged to do its work in a considerable number of revolutions by means of worm gearing.

Types (1) and (3) would appear to have the advantage of greater adaptability to varying pressures. It is an easy matter

to gear the motor down so that it will turn under the lowest air pressures anticipated. At higher pressures wire drawing at the ports prevents the attainment of an undesirably high speed.

In type (2) hydraulic leather packings are used, and considerable care should be taken in the design and the workmanship to eliminate all unnecessary sources of friction. The strict alignment of the two cylinders and the guided end of the rod deserve special attention.

Literature.—Ford, J. M., "High Pressure Air Compressors." —Paper read before the Greenock Assoc. of Shipbuilders and Engineers. See *Engineering*, October 20th, 1916, *et seq.*

CHAPTER XIII

VALVE GEAR

Cams.—With few exceptions the valves are operated by external profile cams made of cast iron chilled and ground on the face. In order to facilitate the removal of valves and cylinder covers, it is usual to arrange the cam-shaft to one side of the cylinders and to transmit motion from the cams to the valves through levers or a combination of levers and push-rods or links. The arrangements in general use give an approximate one to one leverage between cam and valve, and the figures

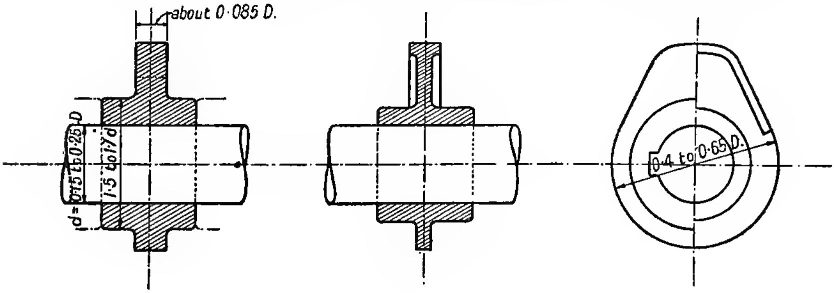


FIG. 225.

given above for the width of cam face are based on this proportion.

Two forms of cam body for the air and exhaust valves of four stroke non-reversible engines are illustrated in Fig. 225, the dimensions being expressed in terms of the cylinder bore. Fig. 226 shows a combined ahead and astern cam for a large marine engine. The bosses should be bored a hard-driving fit on the cam-shaft, and their lengths should be machined accurately to dimensions, so that the complete group of cams required for one cylinder give correct spacing when driven hard up side by side.

The fuel cam has to be of special construction, on account of

the necessity for precise adjustment of the timing, and a typical form is shown in Fig. 227. The toe-piece is preferably made of hardened steel, but chilled cast iron is sometimes used.

Profile of Cams.—The design of cam profiles for air, exhaust and scavenge values is a matter of reconciling the claims of the following desiderata :—

(1) Rapid and sustained opening.
(2) Absence of wear and noise.

In slow speed engines the question of noise hardly arises, and wear is easily kept to a reasonable minimum by adequate width of face. For such engines the tangent cam shown in Fig. 228 is suitable. For high speeds a smoother shape, as shown in Fig. 229, is desirable, and the relatively slow opening may be compensated by earlier timing. Such profiles are easily drawn by deciding on some arbitrary smooth curve of roller lift, and plotting corresponding positions of the roller with respect to the cam, as in Fig. 230. Some designers are in favour of a sinusoidal form of roller-lift curve. With these smooth profiles peripheral cam speeds of five feet or more per second can be used with quite sweet running. On theoretical grounds the cam profile should be based on the roller clearance circle, as in Fig. 231, but it does not yet appear quite clear whether the procedure has the practical advantages claimed for it.

The starting air cams are best given a sudden rise on the opening side to minimise wire-drawing (Fig. 232).

Combustion valve cam profiles are a study in themselves, and the final decision rests with the test-bed engineers. The effective period is usually about 48 or 50 crank-shaft degrees, or 24 to 25 cam-shaft degrees. The cam-piece should, however, give a range about 25% in excess of this, after allowing for the normal roller clearance to allow for lost motion in the gear. The tangent profile shown in Fig. 227 is usually found quite satisfactory, but other shapes are used.

In order to avoid noise in two stroke engines, it appears necessary either :—

(1) To make the cams of smaller diameter than those of a four stroke engine of the same size, or
(2) To provide double-faced cams mounted on a half-speed cam-shaft.

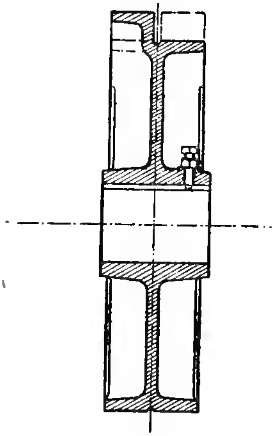


FIG. 226.

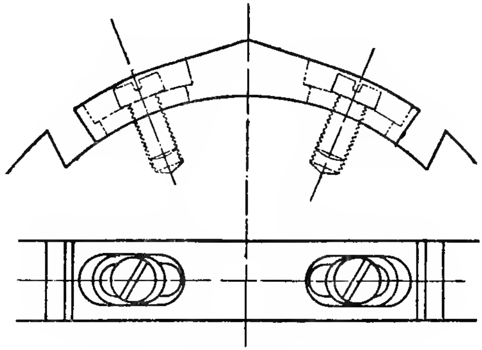


FIG. 227.

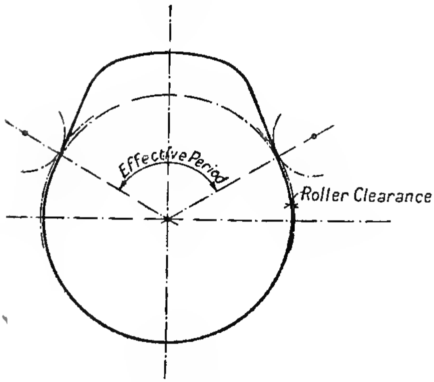


FIG. 228.

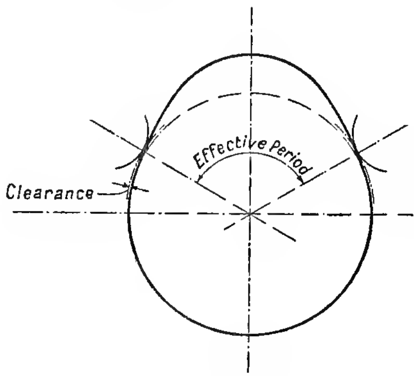


FIG. 229.

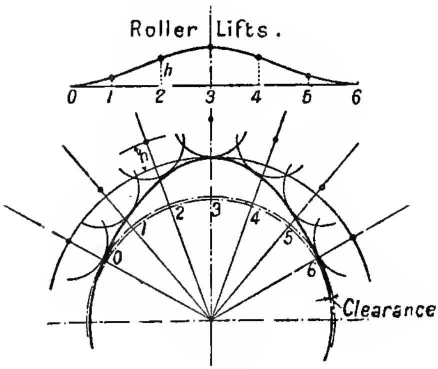


FIG. 230.

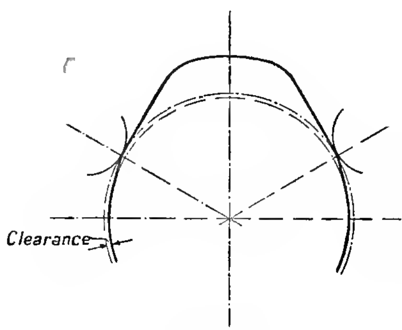


FIG. 231.

The first procedure is the more usual, but the second would appear to have much in its favour. In one marine design advantage is taken of this arrangement to work the engine on the four stroke cycle at slow speeds.

Cam Rollers.—Engines having longitudinally fixed camshafts are usually provided with cam rollers of steel case-

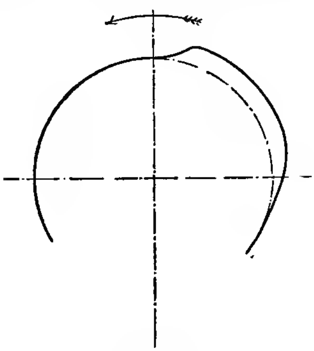


FIG. 232.

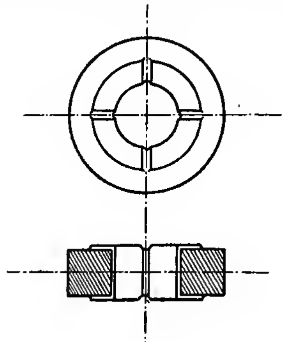


FIG. 233.

hardened and ground inside and out (Fig. 233) and having a diameter of about one-third that of the corresponding cams. The grooves provided for hand lubrication of the pin should be noted. In marine engines in which reversal of rotation is effected by the provision of ahead and astern cams mounted on a longitudinally movable shaft, the rollers require to be of large diameter (about 60% of the cam diameter), in order that the idle cam may clear the lever, as in Fig. 234. Rollers of this size may be of cast iron bushed with phosphor bronze.

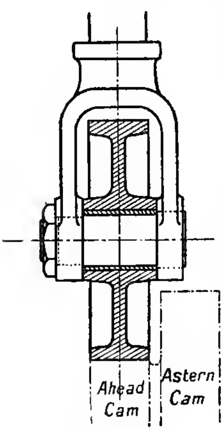


FIG. 234.

Valve Levers.—A common arrangement of valve levers and lever fulcrum shaft for four stroke land engines is shown in Fig. 235. The fulcrum brackets are secured to the cylinder cover, and the latter may be lifted complete, with all valves and gear, and replaced without disturbing the valve settings. With the arrangement shown it is necessary to lift away the fulcrum shaft and levers before the various valves can be removed for regrinding. This very slight inconvenience

is sometimes overcome by means of split levers or by provision of horse-shoe shaped distance collars on the fulcrum shaft, which when removed leave sufficient space to allow the levers to be moved sideways clear of the valve casings. These devices are desirable in the largest engines only. Referring to Fig. 235 below, it will be noted that the fuel and starting levers are mounted on an eccentric bush A, connected to the handle B. The latter is provided with a spring catch engaging with notches in the fixed disc C, in accordance with the following scheme and the diagram shown in Fig. 236.

Top notch.—Running.—Fuel lever in its normal running position. Starting valve roller out of range of cam.

Middle notch.—Neutral.—Both fuel and starting valve rollers out of range of cams.

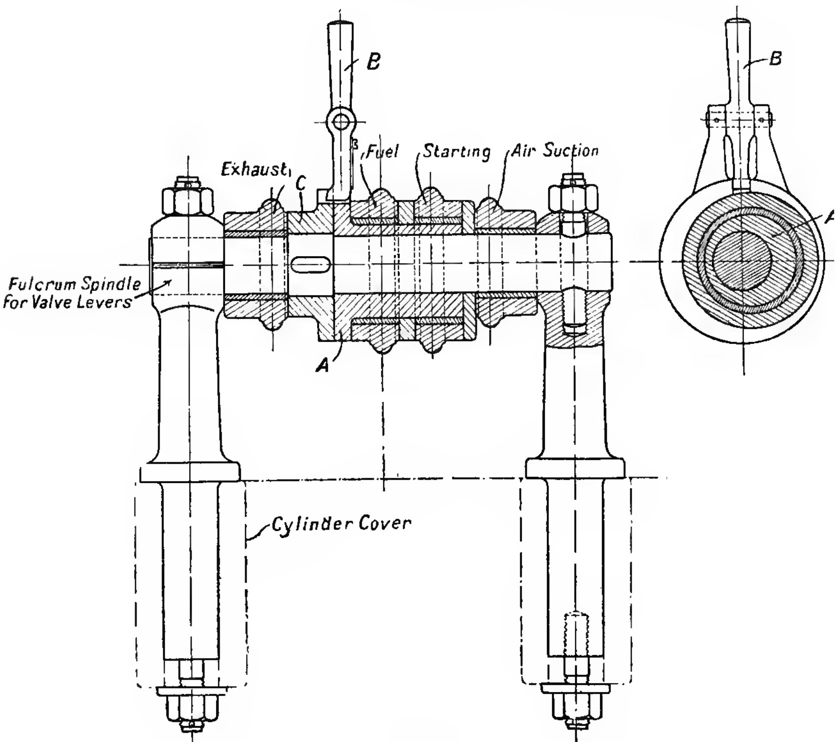


FIG. 235.

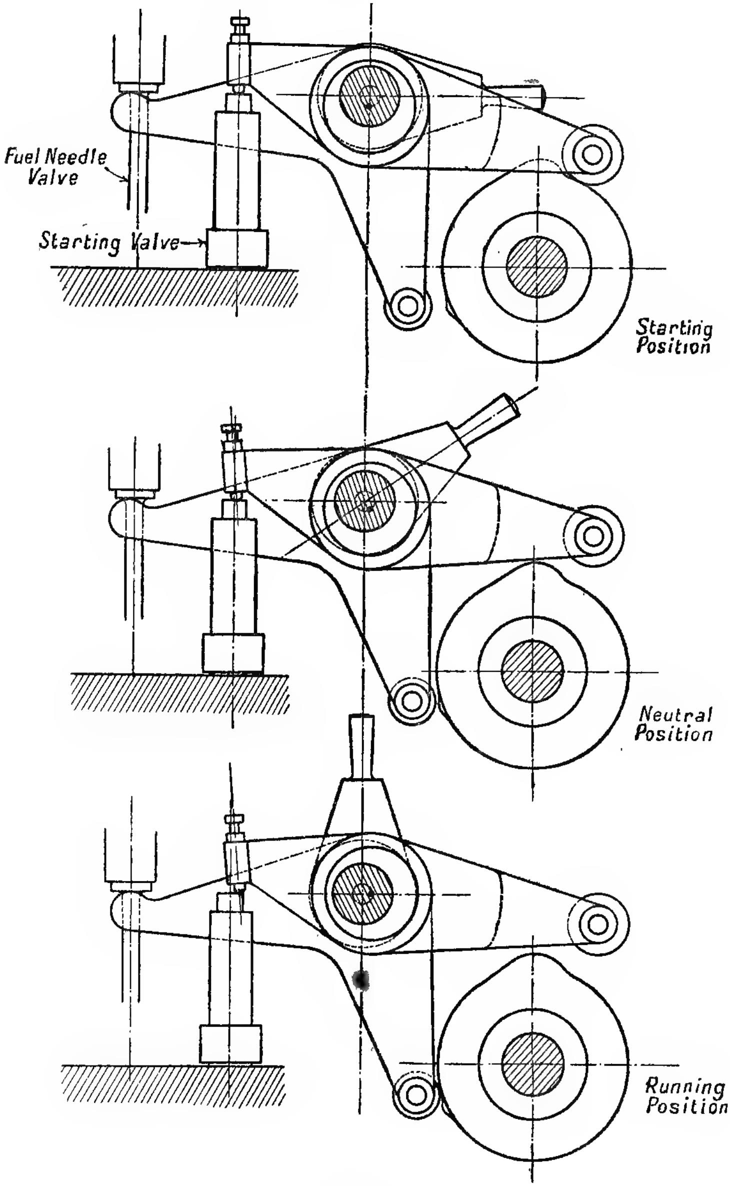

Fuel Needle Valve
Starting Valve
Starting Position
Neutral Position
Running Position

FIG. 236.

Bottom notch.—Starting.—Starting air valve lever in its working position. Fuel valve roller out of range of cam.

Sometimes this arrangement is modified by keying the eccentric bush and handle to the fulcrum shaft and allowing the latter to turn in its supports. This scheme is useful when the disposition of the gear is such that an air suction or exhaust lever separates the fuel and starting levers. This eccentric mounting of levers may also be used for exhaust lifting or to remove all the levers out of range of the cams during the axial displacement of the cam-shaft of a reversing engine.

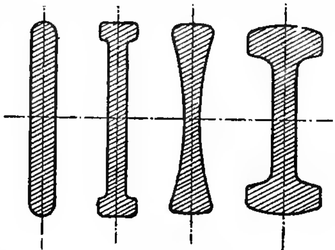


Fig. 237.

Certain well-known marine makers of great repute do not consider it necessary to put the fuel valve out of action whilst the engine is running on compressed air, and content themselves with suspending the supply of fuel to the valves during this period.

The levers are generally of cast steel or malleable iron, but good cast iron may be used if the stress is confined to about

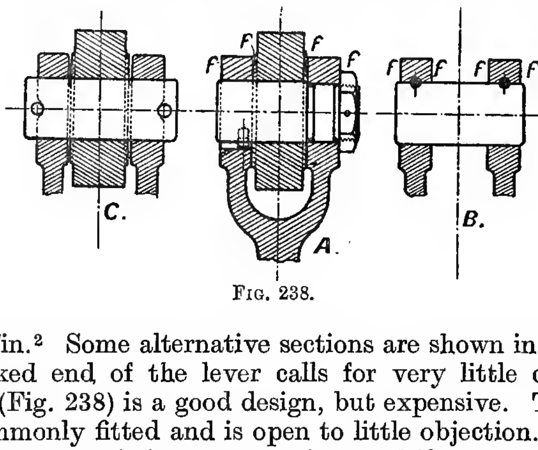


Fig. 238.

2000 lb./in.² Some alternative sections are shown in Fig. 237. The forked end of the lever calls for very little comment. Type A (Fig. 238) is a good design, but expensive. Type B is very commonly fitted and is open to little objection. Type C is the cheapest existing construction, and if accurate castings (machine moulded) are obtainable the only machining opera-

tion required is to drill and reamer the hole for the roller-pin. The two grooves for the taper-pin may be cast.

In small engines the tappet-end may consist of a plain boss screwed to receive a hardened tappet-screw and lock-nut, as in Fig. 239. In larger engines the more elaborate arrangements shown in Fig. 240 are usually adopted. The bosses of the levers should be bushed with good phosphor bronze and provided with a dustproof oil cup of some description, in order to reduce wear to a minimum. With these precautions the bush should only require renewal at widely distant intervals

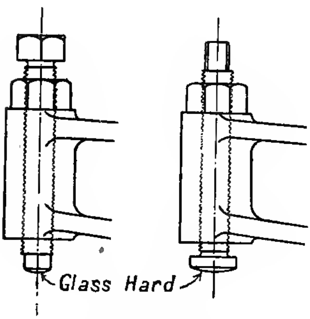


Fig. 239.

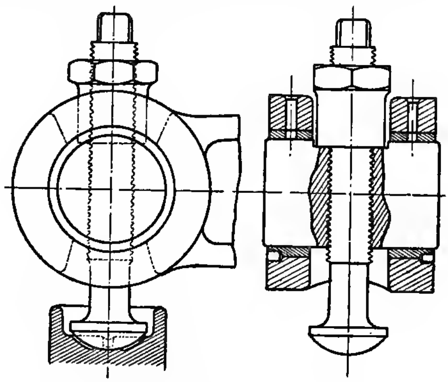


Fig. 240.

and means of adjustment are unnecessary even in the largest sizes of engines.

Strength of Valve Levers.—In four stroke engines the exhaust-valve lever is the most heavily loaded. Although the force required to operate the suction valve is relatively small, it is usual to make the inlet valve lever of the same section as the exhaust lever for the sake of uniformity of appearance, and the same pattern may frequently be used for both. The loads imposed on the tappet-ends of the various levers at the points of valve-opening are given below :—

Four Stroke Engines

Exhaust Valve. About 45 lb. per sq. in. of exhaust valve area + spring load + inertia of valve.

Suction Valve. Spring load + inertia of valve + a maximum of about 5 lb. per sq. in. of valve area if the exhaust valve happens to be closing too early, due to excessive roller clearance.

Starting Valve. 500 lb. per sq. in. of valve area + spring load.
Fuel Valve (a) Swedish type.
1000 lb. per sq. in. of needle area + spring load + inertia, all reduced by the leverage employed.
(b) Augsburg type.
Difference between the spring load and 1000 lb. per sq. in. of needle area at stuffing-box.

Two Stroke Engines

Scavenge Valves. Spring load less scavenge air pressure into area of valve + inertia of valve.
Fuel and Starting Valves. As for four stroke engines.

All the above are of course subject to slight correction for friction.

By way of example, the main dimensions of the exhaust valve lever for a 20″ four stroke cylinder are calculated below :—

Data :

Diameter of exhaust valve, 6·5 in.
Fulcrum spindle and exhaust valve lever centres, as in Fig. 241.
Pressure load on exhaust valve
$=0{\cdot}785\times6{\cdot}5^2\times45=1490$ lb.
Spring load at, say, 8 lb. per in.^{2} of valve area
$=0{\cdot}785\times6{\cdot}5^2\times8=$ 265 lb.
Inertia load, say . . 40 lb.
Total load to open exhaust valve . . 1795 lb.

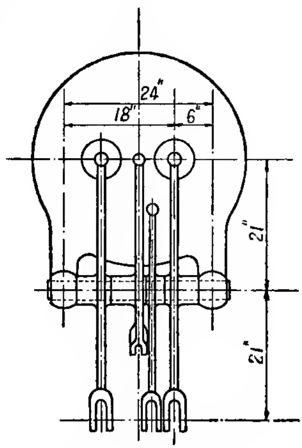


Fig. 241.

Reaction at fulcrum spindle, about 3600 lb.

Bending moment at fulcrum spindle $=\dfrac{3600\times6\times18}{24}$ in. lb.

Allowing a stress of 6000 lb./in.^{2},

$$\frac{d^3}{10}=\frac{3600\times6\times18}{6000\times24}=2{\cdot}7$$

$$d=3 \text{ in.}$$

Allowing for a bush $\frac{1}{4}''$ thick and about $\frac{3}{4}''$ metal at the boss the external diameter of the latter will be 5″.

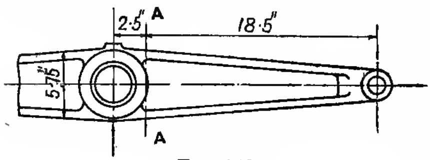


Fig. 242.

Sketching in the approximate outline of the lever, as in Fig. 242, it is seen that at the weakest section AA, the bending moment is about $1800 \times 18{\cdot}5$ in. lb., and taking a stress of 5000 lb./in. for cast steel, the modulus Z of the section AA should be $\frac{1800 \times 18{\cdot}5}{5000} = 6{\cdot}66$ in.^{3}

If the section AA is approximately T-shaped, as in Fig. 243, then Z=b.t.h. nearly. "h" is 5·75, and therefore

$$b.t = \frac{6{\cdot}66}{5{\cdot}75} = 1{\cdot}16 \text{ in.}^2$$

which is satisfied by $b = 2{\cdot}25$ and $t = 0{\cdot}515''$.

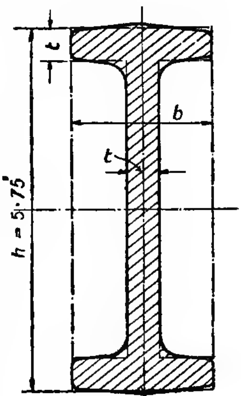


Fig. 243.

The lever may be made of approximately uniform strength by tapering towards the ends, both in width and depth, as in Fig. 244, whilst the flange and web thicknesses are kept constant. If a double bulb or other section is required, for the sake of appearance, and on casting considerations, it is a simple matter to sketch in such a section approximately equivalent to the simple I-section to which the calculation applies.

Push-rods.—In some designs a push-rod is introduced between the lever and the cam-roller, as shown diagrammatically in Fig. 199 *ante*, in order to enable the cam-shaft to be located at a low level. For this purpose bright hollow shafting, or even black lap-welded steam tubes are suitable, if not too highly stressed.

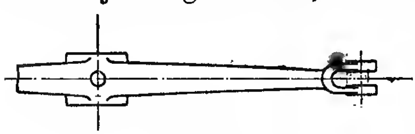


Fig. 244.

For handy reference in designing such push-rods the following table, taken from Prof. Goodman's "Mechanics Applied to Engineering," is given for the buckling loads of tubular struts. In using these figures it is advisable to use a factor of safety of

not less than about 3 or 4, and in no case to employ stresses exceeding 10,000 lb./in.^{2}

BUCKLING STRESS (FREE ENDS), LB. PER SQ. IN.

Ratio $\frac{\text{length.}}{\text{diameter.}}$	Mild Steel.
10	59,000
20	42,000
30	29,000
40	20,000
50	14,000
60	10,500
70	8,200
80	6,500
90	5,500
100	4,500

The jointed ends of the rods may be of forged steel bar or malleable cast iron bushed with bronze, as in Fig. 245.

Cam-shafts.—In modern shops the cam-shaft may be rapidly and cheaply ground to size from black bars. In order to facilitate the driving on of the cams for a multi-cylinder engine the enlarged diameters are usually made of increasing sizes, differing by successive thirty-seconds of an inch, or thereabouts, as shown exaggerated in Fig. 246. The same figure which represents the cam-shaft for a four stroke generating set of three cylinders also shows the method of supporting the shaft by means of a continuous trough with one bearing between each bank of cams. This arrangement has a very neat appearance and makes provision for catching the oil which drips off the cams and rollers. In some designs the cams are allowed to dip into an oil-bath, the level of which is maintained constant by a small pump

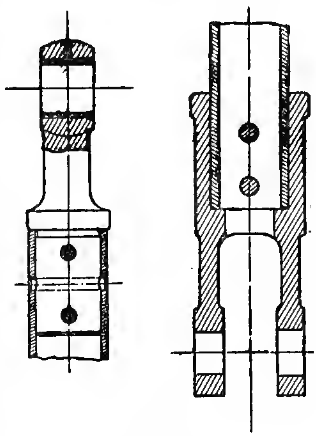


FIG. 245.

provided for the purpose, or by a connection taken from the forced lubrication system. A copious supply of oil to the cams has the advantage of securing quiet running.

In other designs the cam-shaft is supported by bearing brackets secured to the cylinders as in Fig. 247. In this case

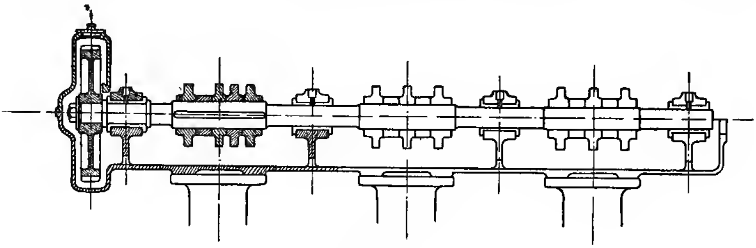


Fig. 246.

it is very desirable to fit light cast or sheet iron guards round each bank of cams. The cam-shaft bearings are divided horizontally for adjustment, and the shells may be of cast iron lined with white-metal, solid gun-metal, or in small engines, where the cost of material does not outweigh the advantage of simplicity, of solid die-cast white-metal. Owing to the slow peripheral speed and the intermittent character of the loading,

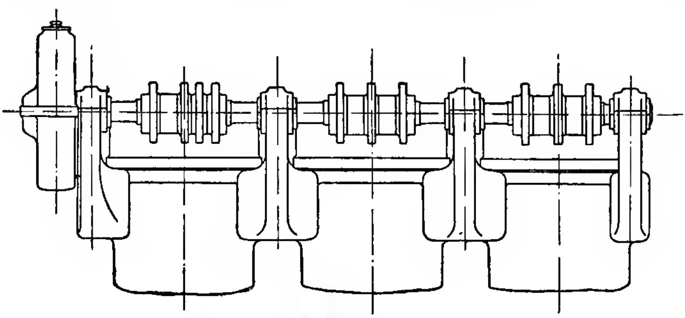


Fig. 247.

grease lubrication by Stauffer boxes is quite adequate, although ring and syphon are more commonly used.

A slightly different arrangement is shown in Fig. 248. Here the shaft is supported by a series of cam-troughs, one to each cylinder, each trough having two bearings. The extra rigidity of this arrangement allows of the cam-shaft diameter being reduced below the figure required with the other arrange-

ments described. This division of the trough into segments is advantageous from the manufacturing point of view as the smaller parts are easier to cast and handle in the shops; also one pattern serves for engines of any desired number of cylinders.

Strength of Cam-shafts.—The size of cam-shaft required for a given engine would appear to depend not so much on the stresses to which it will be subject, as on the rigidity necessary to secure sweet running of the gear. For a four stroke engine the opening of the exhaust valve against the terminal pressure in the cylinder is the severest duty which the cam-shaft is called

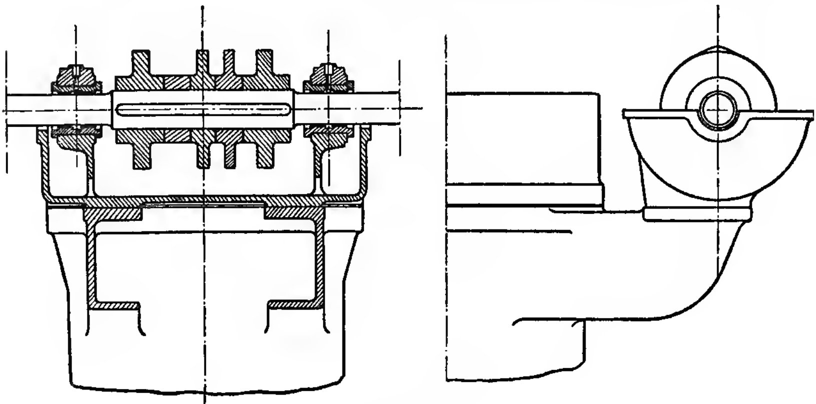


FIG. 248.

upon to perform. The load is applied and released fairly suddenly, and a cam-shaft lacking in torsional and transverse rigidity would undoubtedly be subject to oscillations, which in an acute case would give rise to the following evils:—

(1) Noisy action of cams, due to torsional recoil of shaft after each exhaust lift.

(2) Interference with the timing of valves (particularly the fuel valves) of cylinders remote from the gearing end of the cam-shaft.

(3) Chattering of the gear-wheels by which the shaft is driven.

In view of the fact that as shaft diameters are increased the stiffness increases at a greater rate than the strength, it seems just possible that strength considerations might outweigh those of stiffness in very large engines. On the other hand,

if angular deflection of the shaft between contiguous cylinders be accepted as the criterion, then considerations of similitude give shafts of diameters bearing a constant ratio to the cylinder bores (or rather exhaust valve diameters) and constant stresses in all sizes if the terminal pressure is always the same. The fact that in practice relatively thinner cam-shafts are used in large engines may be due to the lower terminal pressures obtaining in the cylinders of the latter.

The following table shows the approximate diameters of cam-shafts used in practice on four stroke engines of different sizes :—

Bore of Cylinder in inches	6	10	15	20	25	30
Diameter of Cam-shaft in inches	$1\frac{1}{2}$	$2\frac{1}{4}$	$2\frac{7}{8}$	$3\frac{3}{8}$	$3\frac{7}{8}$	$4\frac{1}{4}$

The above figures hold for any number of cylinders up to four with the cam-shaft drive at one end, or eight with the cam-shaft drive at the centre.

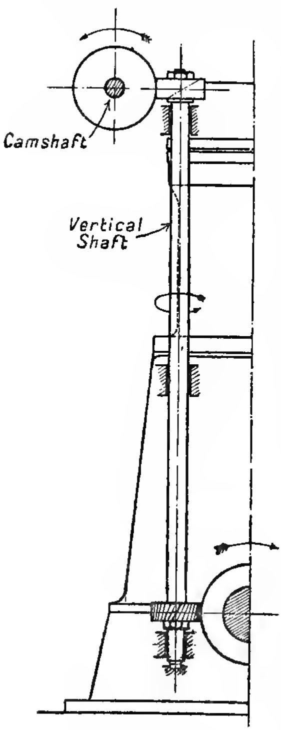


FIG. 249.

For two stroke engines these diameters may be materially reduced on account of the absence of exhaust valves. Average figures for existing practice appear to be about 25% lower than those given above for four stroke engines. The fuel pumps and cylinder lubricating pumps are frequently driven off the cam-shaft, but any auxiliary gear, such as circulating pumps, etc., requiring appreciable power are precluded.

Cam-shaft Drives.—For non-reversible engines, the spiral drive shown diagrammatically in Fig. 249 is the favourite. This drive comprises the following components :—

(1) Lower spiral wheels.
(2) Footstep bearing for vertical shaft.
(3) Vertical shaft and couplings.
(4) Upper spiral wheels.

The lower spiral wheels generally have a 1 : 1 ratio, so that the vertical shaft runs at engine speed. In some designs the ratio is $1\frac{1}{2}$: 1, and the vertical shaft runs

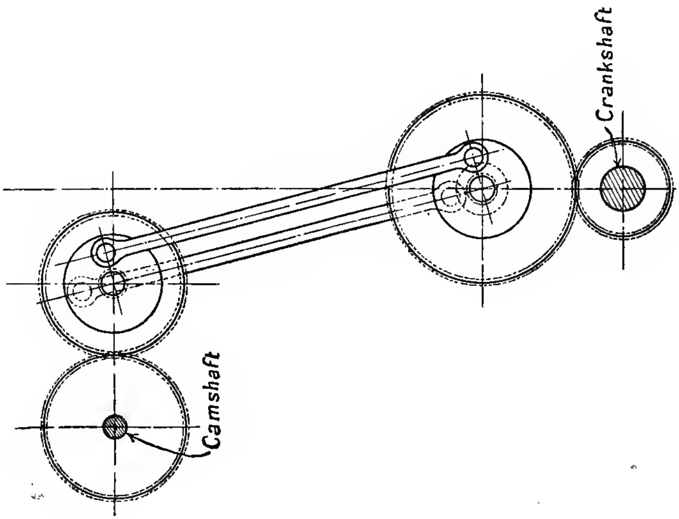


FIG. 253.

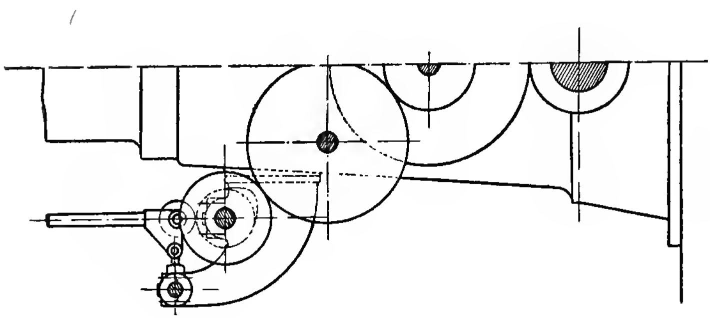


FIG. 252.

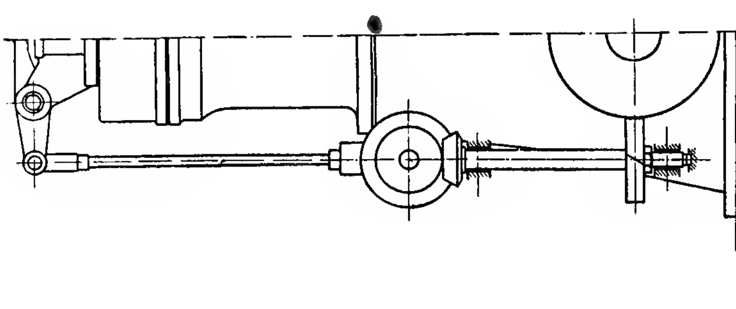


FIG. 251.

cam-shaft, due to spiral wheels, etc., preferably by ball-thrust washers.

(3) The wheels to run in a bath of oil, and suitable arrangements to be made to prevent leakage of the latter.

(4) The general arrangement of gear-box and bearings to be compact and in general conformity with the design of the rest of the engine.

Probably the simplest way of fulfilling the above requirements is to cast the gear-case *en bloc*, with a continuous cam-trough, as in Fig. 246 *ante*, or in the case of a central drive to suspend the gear-case between two sectional troughs, as in Fig. 254, by a sufficient number of fitted bolts.

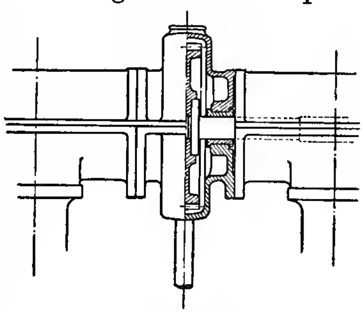


Fig. 254.

Spur and Spiral Gears for Cam-shaft Drives.—The question of the strength of the teeth hardly arises in this case, and the problem consists in the selection of materials and proportions giving quiet running and absence of wear. The following pairs of materials are in common use :—

	Driver.	*Follower.*
(1)	Cast iron.	Cast iron.
(2)	Steel.	Cast iron.
(3)	Steel.	Bronze.

Of these, the pairs (1) and (2) appear to give the best results, with proper proportions and adequate lubrication, etc.

In good practice, the normal circular pitch of the teeth is made about equal to one-twelfth of the cylinder bore for both spur and spiral gears, and the width of face about one-fifth of the cylinder bore in the case of four stroke engines. It is a fairly safe rule to make the pitch as coarse as the smallest wheel will allow in the case of spiral wheels. With spur wheels fine pitches are not so objectionable as the sliding between the teeth is much less.

For satisfactory running, the teeth must of course be properly cut and the wheels accurately centred. The tooth clearance should not exceed about 2/1000″, and should be uniform all round.

For particulars of tooth-gearing calculations the reader is referred to the special books devoted to this subject. The diagram shown in Fig. 255 is very useful in the preliminary stages of spiral drive calculation. Suppose it is desired to design a pair of right-angle spiral wheels of say 1 : 2 ratio; first calculate the diameters of a pair of spur gears of the desired pitch and giving the desired ratio, viz., 1 : 2. Draw OA and OB equal to the pitch radii of the follower and driver respectively. Complete the rectangle OBCA and draw any line DCE, cutting the axes in D and E. Then DC and CE will be equal to the pitch radii of equivalent spiral wheels having

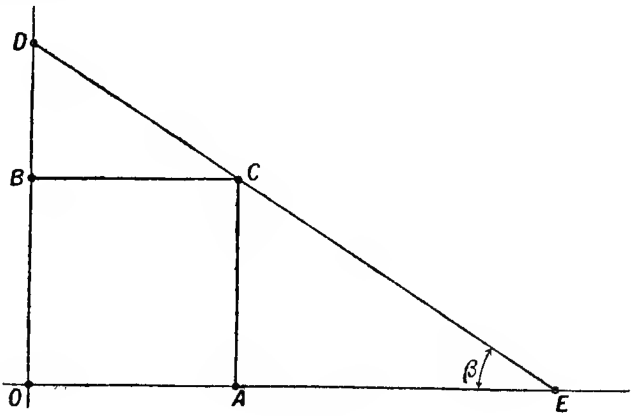


FIG. 255.

spiral angles α and β and having a normal pitch the same as the circular pitch of the spur wheels first calculated. It usually happens that the wheel centres (DE) are fixed within approximate limits by space considerations, and a process of trial and error is required to find suitable values for the number of teeth and the spiral angles. The latter should not be less than about 27°, as the efficiency falls off rapidly as this figure is reduced. Having obtained an approximate solution by the above method, the angles should be determined to the nearest minute by logarithmic trial and error calculation by means of the following relation :—

$$\frac{AC}{\sin \beta}+\frac{BC}{\cos \beta}=DE=\text{required wheel centres.}$$

Reversing Gears.—In spite of early anticipations of difficulty, reversing gears for Marine Diesel Engines have attained a high degree of efficiency. On the score of simplicity, reliability and quick action they compare favourably with the corresponding parts of steam engines. A very great number of different gears have been suggested and patented, but those in widespread use fall into two or three well-defined classes, which will be described below.

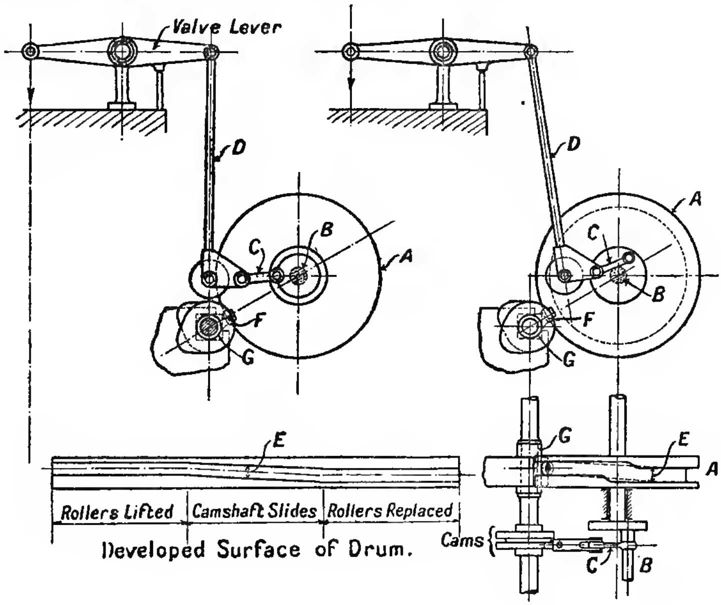


FIG. 256.

Sliding Cam-shaft Type of Reversing Gear.—This type of gear is the favourite for four stroke engines, though it is equally applicable to those working on the two stroke cycle. Ahead and astern cams side by side are provided for the operation of each valve. Reversal is effected by sliding the cam-shaft a few inches endways in its bearings, so that the ahead cam is removed from the action of the roller and replaced by the astern cam and vice versa. It is in general necessary to arrange means whereby the valve rollers may be swung clear of the cam noses during the longitudinal movement of the shaft, otherwise fouls would occur. In some very small

engines the necessity for such provision is obviated by employing curved-faced rollers adapted to slide up and down inclined faces between the ahead and astern cams respectively.

The method adopted in some of the Burmeister & Wain engines is shown in Fig. 256. A drum A is mounted on a cranked shaft B, on which are hinged drag links C, connected to the roller end of the valve push-rods D. Drum A is provided with a groove E, the developed shape of which is shown in the figure. This groove accommodates a roller F, attached to a movable collar bearing G. Shaft B is rotated in the direction desired ("ahead to astern" or "astern to ahead") by

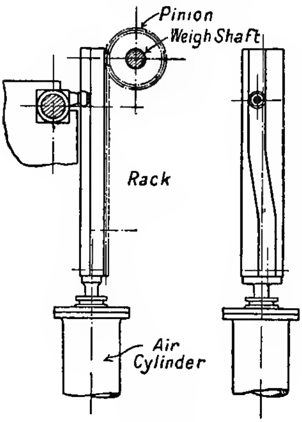


Fig. 257.

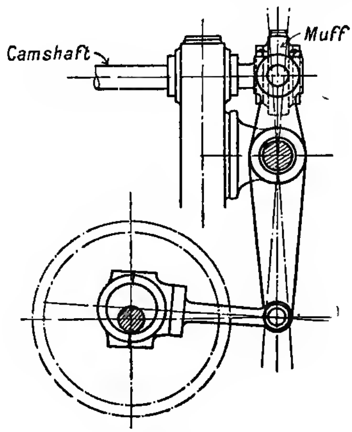


Fig. 258.

suitable gearing in connection with a reversing servo-motor or the like. Approximately one-third of a revolution of the shaft suffices to swing the rollers clear of the cams; meanwhile the cam-shaft is stationary. Another approximate one-third of a revolution causes the groove E to shift the cam-shaft from ahead to astern positions, or vice versa, whilst the valve rollers execute a harmless movement a little further out and back again. The remainder of the revolution of the weigh-shaft B replaces the rollers in their running position.

In other engines of the same make, the developed shape of the groove E is executed on the back of a rack by means of which the straight line motion of a vertical servo-motor is converted into rotary motion of the weigh-shaft. This variation is shown in Fig. 257.

If separate means be adopted for removing and replacing the rollers, it is obviously possible to devise very simple means, of shifting the cam-shaft as in Fig. 258 for example. In such cases the two mechanisms must be interlocked to prevent a false manœuvre.

Experiments show that the force required to move the cam-shaft longitudinally is about one-third of the weight of the cam-shaft, plus cams and other gear keyed thereto, and this figure may be used as a basis of calculation for this type of gear. It is advisable, however, to allow a fair margin of power, as the resistance to motion must always be a matter of some uncertainty. When the axial motion of the shaft has the

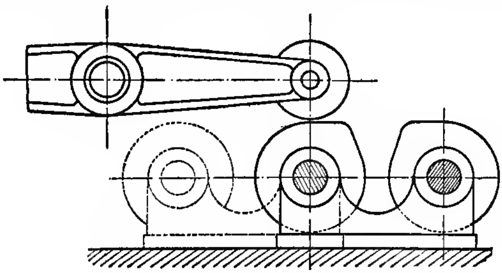


Fig. 259.

effect of opening one or more of the valves, the resistance due to this cause must be added to that of the shaft itself.

Twin Cam-shaft Type of Reversing Gear.—With this type of gear, which is a speciality of the Werkspoor Company, not only are separate cams provided for ahead and astern running, but the latter are mounted on separate cam-shafts capable of being slid into and out of action as required. Fig. 259 illustrates the arrangement diagrammatically. The cam-shaft drive is usually by means of coupling rods. The chief advantage of this gear would appear to be the absence of special gear for swinging the rollers out of operation, this process being unnecessary. It is perhaps worth noticing here that the simplicity of the manœuvring gear is one of the outstanding features of the Werkspoor Marine Diesel Engine.

Twin Roller Type of Reversing Gear.—This gear depends on some form of link-work such as that shown in Fig. 260. Rollers A and B lie in the planes of the ahead and astern cams respectively. In the position shown the timing of the valve

is controlled by the ahead cam and roller A; roller B is meanwhile outside the radius of action of its cam. Rotation of the weigh-shaft C through a predetermined angle throws roller A out of action and brings roller B into action with the astern cam. This type of gear has been applied to both four and two stroke engines.

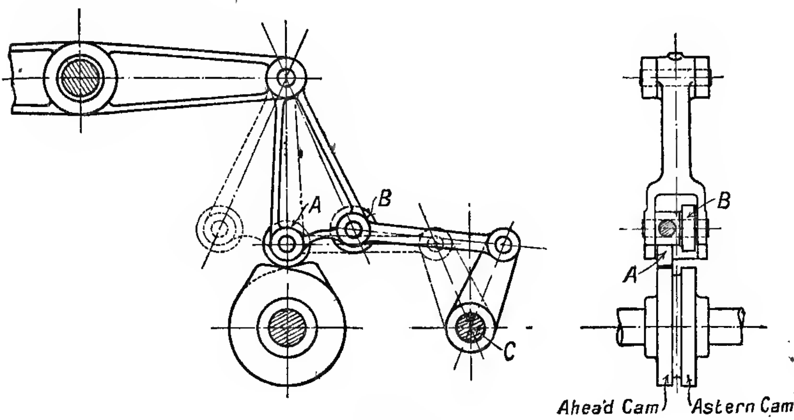


FIG. 260.

A different arrangement, having some slight resemblance to the above, is shown in Fig. 261. In this case there is only one roller which is swung from the ahead to the astern cam by a motion in a plane at right angles to the plane of the gear. The roller face is curved to allow of this slight angular displacement from the vertical. The inherent defects of this mechanism probably render it unsuitable for use in conjunction with any but the air starting valves.

Selective Wedge Type of Reversing Gear.—This ingenious gear, illustrated diagrammatically in Fig. 262, has been devised by Carels Frères and used in connection with the starting air and fuel valves of two stroke marine engines designed by them. Ahead and astern cams are provided side by side, and the valve roller A is wide enough to cover both. Between the cams and the lever is interposed a roller-wedge piece B, under control of a cam C, mounted on a manœuvring shaft D. The latter is capable of independent rotary and endway motion. A suitable rotary motion of the shaft D withdraws the wedge B to an extent which renders inoperative the ahead cam on which it rests. A longitudinal movement of shaft D carries the

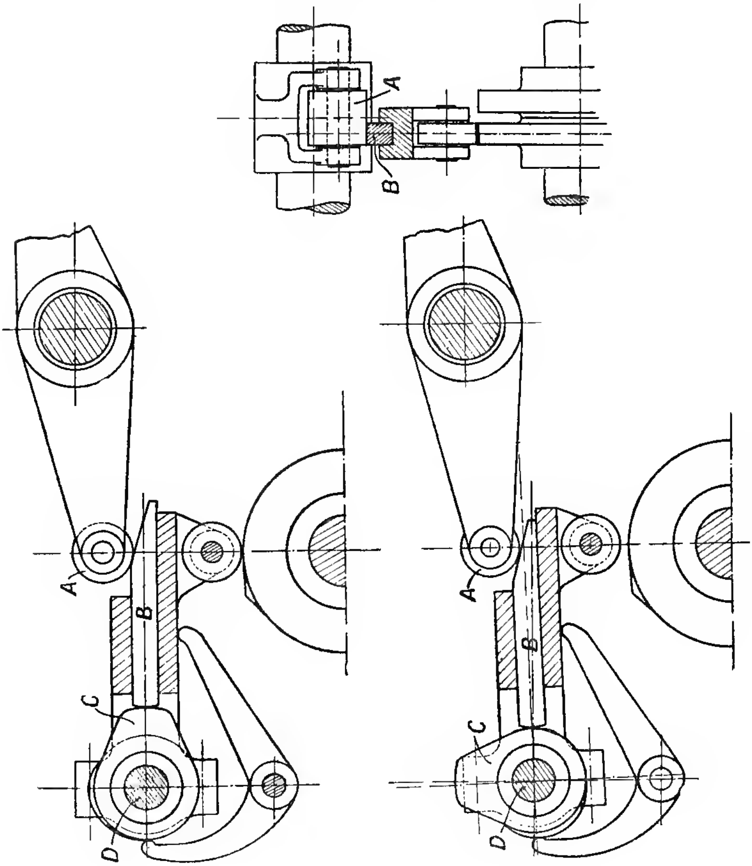


FIG. 262.

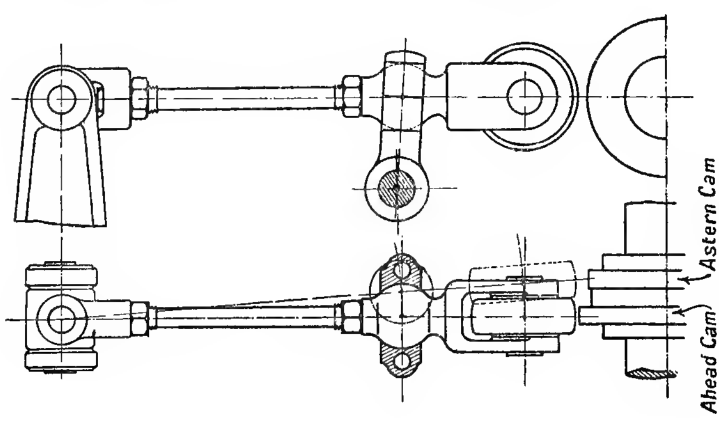


FIG. 261.

wedge B with it, and a further rotation of D introduces the wedge between roller A and the astern cam, and vice versa for astern to ahead. It is to be noticed that by a suitable arrangement of the durations and sequences of the cams by which the wedges are operated the engine is caused to start up in any predetermined manner, as for example :—

Position (1) Six cylinders on air. Fuel valves inoperative.
,, (2) Three cylinders on air. Three cylinders on fuel.
,, (3) Six cylinders on fuel. Air valves inoperative.

Special Reversing Gears for Two Stroke Engines.—With two stroke engines there are a number of means by which the duplication of cams may be avoided. Considering the case of

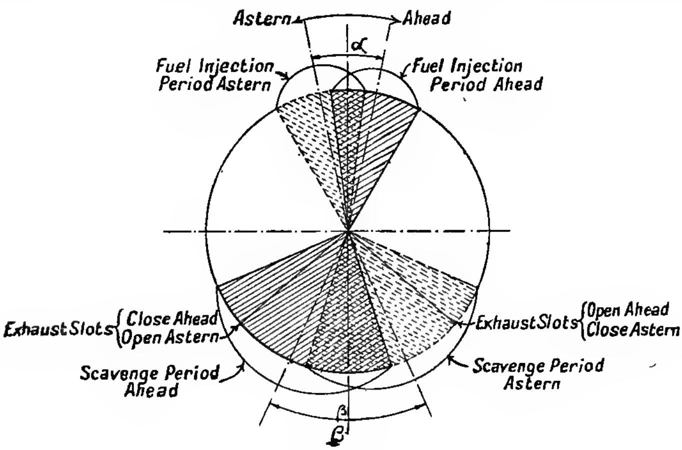


Fig. 263.

an engine fitted with scavenge valves and neglecting the starting air valves for the moment, the valve settings for ahead and astern will be somewhat as shown in Fig. 263. It will be seen that for both fuel and scavenge valves all that is required to effect reversal is the rotation of the cam-shaft through a certain angle α for the fuel valve and β for the scavenge valves. In some early engines it was decided to select $\alpha=\beta=$about 30° to 35°, and so effect reversal of both valves by one movement. In later engines, however, it is more usual to use the rotation of the cam-shaft to reverse the scavenge valve only and adopt independent means, such as duplicate cams, etc., for the fuel and starting valves. The effect of the rotation of the cam-

shaft on the settings of the latter must of course be allowed for in fixing the angular positions of the fuel and starting cams.

A simple method of effecting the desired rotation of the cam-shaft of a small engine is shown diagrammatically in Fig. 264. Spiral drives are used and the vertical shaft is in two pieces, C and D, connected by a splined coupling permitting vertical movement of the upper half D. The vertical

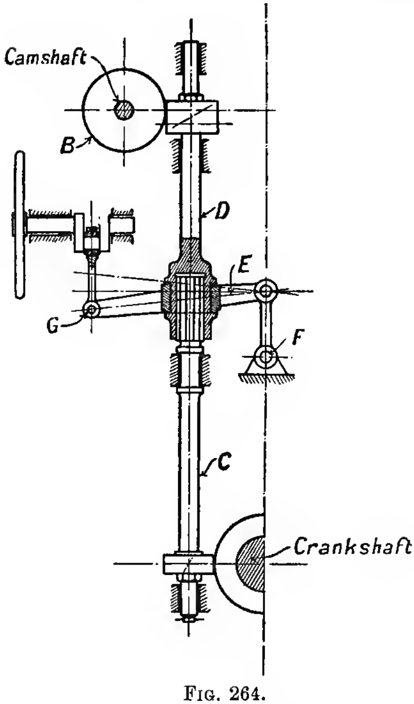


FIG. 264.

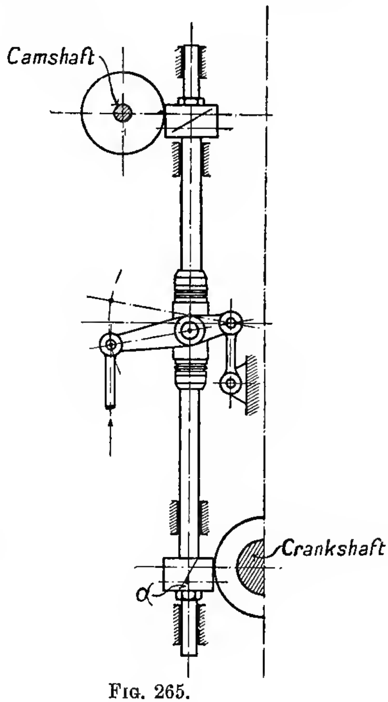


FIG. 265.

position of D is determined by the lever E, which is hinged at F and connected at G to an eccentric or other suitable means of transmitting motion from the hand-wheel. The extreme upper and lower positions of D determine the ahead and astern running positions.

If h = Lift of vertical shaft,
r = Pitch radius of wheel B,

Then $\frac{h}{r}$ = Reversing angle in radians.

In other arrangements the vertical shaft is moved as a whole, as in Fig. 265, and in calculating the amount of motion required for a given reversing angle it is necessary to take into account the rotation of the vertical shaft due to the sliding between the lower helical wheels.

Consider the case where both upper and lower gears have a ratio of 1 : 1. Let a be the spiral angle of the crank-shaft gear-wheel.

Note that $a < 45°$.

Let h = Lift of vertical shaft.
r_1 = Pitch radius of vertical shaft lower wheel.
r_2 = Pitch radius of cam-shaft wheel.

Then

Rotation of cam-shaft due to axial movement of vertical shaft $= \frac{h}{r_2}$ radians as before.

Further,

Rotation of vertical shaft due to sliding of lower spiral wheels $= \frac{h.\tan a}{r_1}$

Therefore,

$$\text{Reversing angle} = \frac{h}{r_2} \pm \frac{h \tan a}{r_1}$$

With the arrangement shown the positive sign applies when the upper and lower spirals have the same hand and the negative sign when they are of opposite hand.

An inspection of the valve settings for ahead and astern, as shown in Fig. 263 *ante*, reveals the fact that the " reversing angle " is always described in the direction opposite to the previous direction of motion. Advantage has been taken of this fact to obtain self-reversing valve settings by arranging, between the cam-shaft drive and the cam-shaft proper, a claw clutch having angular clearance between the jaws equal to the reversing angle. With this arrangement independent reversible gearing must be used for the starting air valves. A suggested improvement on the above is to provide mechanical means for taking up the slack between the jaws whilst the engine is standing, instead of allowing it to be suddenly taken up on starting.

Movable Roller Type of Reversing Gears.—Instead of rotating the cam-shaft through a certain angle relative to the

cam-roller, the same effect may be obtained by turning the roller relative to the cam-shaft.

One such arrangement is shown in Fig. 266. It will be noticed that the valve lift is less in the astern position than in the ahead, but this is unimportant. A similar device is shown in Fig. 267. In both these designs the reversing angle is conveniently halved by fitting double-nosed cams to a half-speed cam-shaft.

Another gear coming under this category is shown in Fig. 268.

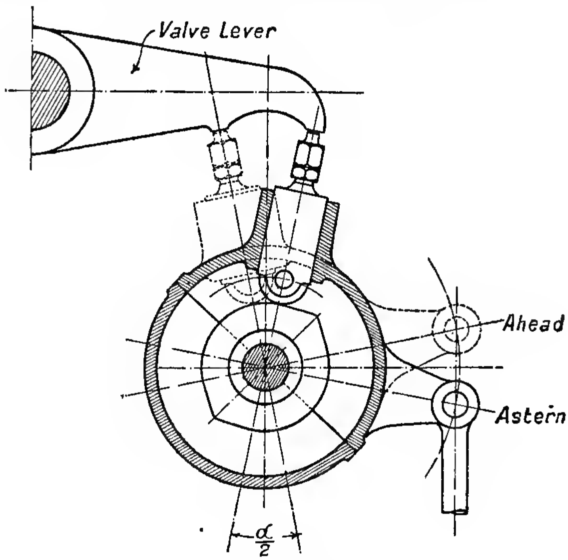


FIG. 266.

The displacement of the roller from its ahead to astern position is effected by the partial rotation of the eccentric fulcrum A, and the roller passes through a neutral position, in which it is outside the radius of operation of the cam.

The above descriptions by no means exhaust the list of existing Diesel Engine reverse gears, and doubtless others remain to be invented. It is evident, therefore, that the problem of reversibility no longer presents any obstacle to the development of the Diesel Engine for marine service.

For slow-running engines there is probably little serious objection to any of the gears which have been described For high speed engines, however, most existing gears for two

stroke engines are noisy and of doubtful durability. Curiously enough, the sliding cam-shaft scheme, which would seem to possess every advantage, does not appear to have been used on this class of engine so far.*

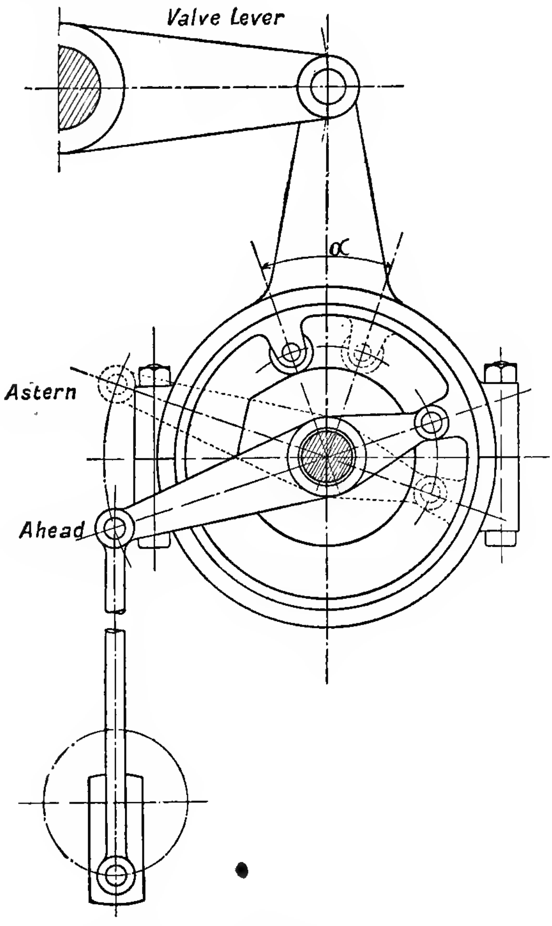


Fig. 267.

Manœuvring Gears.—In this connection the term manœuvring gear is applied to those mechanisms apart from the reversing gear which come into operation on starting up a marine engine.

* This gear has recently been adopted by the firm of Franco Tosi for their two stroke engines.

The procedure differs in different designs, but in general the following remarks are applicable :—

(1) When the engine is standing the blast air supply should be cut off, to prevent accumulation of pressure in any cylinder the fuel valve of which happens to be open. If means be provided for putting the fuel valves out of operation in the stop position the blast cut-out is not so essential, but is still desirable as a safeguard.

(2) The blast air should be turned on automatically immediately the engine is started, although it is quite advantageous to provide an independent shut-off and regulating valve under the control of the engineer.

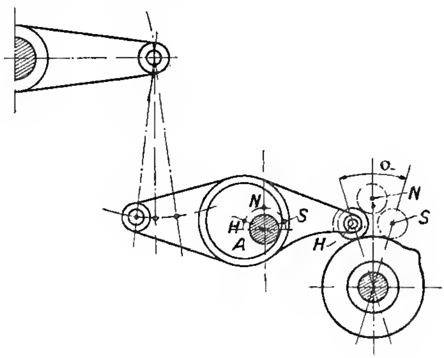


FIG. 268.

(3) When the engine is standing the starting air should be cut off, as there is otherwise great loss of air due to leakage past the starting valves. The starting air shut-off may be automatic or hand operated ; if the latter, it should be opened and closed by one simple motion. An ordinary high pressure globe valve, fitted with a quick-threaded spindle, is suitable for this duty.

(4) The fuel pump suction valves should be held off their seats until such time as the fuel valves are in running position, independent of the position of the fuel control.

(5) The fuel control should be a handle (not a wheel) with a wide range of movement between no oil and full oil.

(6) A wheel, or better still a lever, is provided in connection with suitable mechanism for putting the starting valves into operation at starting, and subsequently putting them out of

operation when sufficient speed has been attained to ensure firing in the cylinders. The same mechanism may or may not (in different designs) throw the fuel valve mechanism out of and into operation. Furthermore, in some designs the operation of this gear is graduated, as in the following scheme, which refers to a six-cylinder engine :—

First notch.—Six cylinders on air (starting).
Second notch.—Three cylinders on air. Three cylinders on fuel.
Third notch.—Six cylinders on fuel.

It has been the experience of many engineers that this graduated control, besides adding to the complication of the gear, is a positive drawback, and that the greatest certainty of starting is secured by passing direct from all cylinders on air to all cylinders on fuel. This arrangement has been adopted on many large four stroke engines with every success, and appears likely to become standard practice. The gear under consideration is connected with the fuel pumps and with the blast starting air supply, so that the following conditions are secured :—

(a) Movement of the lever towards the starting position automatically turns on the starting and blast air.
(b) Suction valves of all fuel pumps held off their seats.

Further movement of the lever puts the starting valves of some or all of the cylinders out of operation, and simultaneously allows normal operation of the corresponding fuel pumps and also of the fuel valves if these latter are arranged to be out of operation during the time the starting valves are working.

(7) Some simple type of interlocking gear is usually fitted to prevent the following false manœuvres :—

(a) Starting the engine before the reversing gear is in either the full ahead or full astern position.
(b) Operating the reversing gear before the manœuvring gear has been put into the stop position.

Some of the means adopted to secure the conditions outlined in sections (1) to (7) will now be described. Further reference need not be made to the reversing gear, as with the exception of the interlocking arrangements mentioned above

the reversing arrangements are entirely independent of the manœuvring gear.

A simple type of manœuvring gear is shown diagrammatically in Fig. 269. The fuel and starting levers are eccentrically mounted on fulcrum shafts, as described earlier in this chapter. Each fulcrum shaft A is connected by links and levers to a manœuvring shaft B, under the control of a hand lever C. In the upper position of lever C all the fuel valves are in operation, and in the lower position the starting air valves. A link D connects the manœuvring lever to an eccentric fulcrum on the fuel pump, by means of which the suction valves are lifted by suitable tappets provided for this purpose, during such time as the starting air valves are in operation. Another link performs a similar operation on the blast air control valve, but in this case the connection is such that the blast air is only cut off in the neutral or stop position of the manœuvring lever.

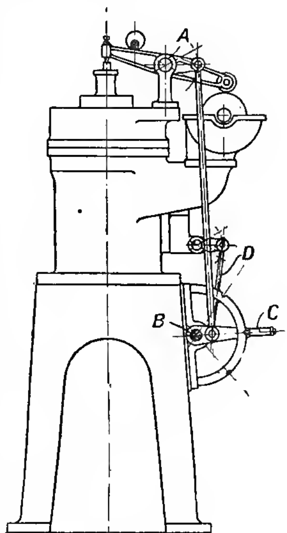


FIG. 269.

In some engines the above arrangements are adopted in principle, but two separate control gears and levers are provided for the forward and aft halves of the engine. The two control levers are placed close together, so that the engineer can work one with either hand. On starting he pulls both towards him, thus putting all cylinders under starting air. As soon as sufficient speed has in his judgment been attained, he pushes one lever towards the fuel position. If firing starts, he then pushes the other lever into the fuel position. If on the other hand firing does not ensue, he may pull back the lever into the starting air position and try the other lever in the fuel notch. With an engine in good order it is probably advantageous to throw over both levers simultaneously.

In other designs it is not necessary to operate the fulcrum shafts, as the starting valves automatically throw themselves into operation when starting air is turned on, and become inoperative when the starting air pressure is released.

A great deal of ingenuity has been expended on the design of gears for throwing successive combinations of cylinders from "air" to "fuel" positions by a continuous movement of a wheel. Some designers have even gone the length of combining the reversing and manœuvring mechanisms so that all positions ahead and astern are secured by clock-wise and anti-clock-wise rotation of this wheel. In the writer's opinion, such gears are not to be desired, for the following reasons :—

(1) Intelligent manipulation of machinery involves a certain parallelism between the mental state of the operator and the response which the machine makes to his control. This state would appear to be most easily secured when separate and distinct operations on the part of the machine are made in response to separate and distinct movements on the part of the operator. Hence it would appear best to keep the reversing and manœuvring control separate, with the exception of whatever measure of interlocking is necessary to prevent accidents.

(2) Gears of the kind referred to are usually complicated, and not easily understood or overhauled.

(3) The complication of such gears is not infrequently associated with backlash, which renders accurate valve setting difficult to effect and maintain.

(4) Complicated gears do not appear to have any practical advantages to offset their increased cost.

Interlocking Gears.—The precise form which an interlocking gear takes in any design depends on the forms of mechanism adopted for the reversing and manœuvring gears respectively, but the problem very frequently reduces to that of two shafts, either of which shall only be capable of movement in prescribed positions of the latter. A simple interlock for two parallel shafts, subject to partial rotation, is shown in Fig. 270. It will be observed that the manœuvring shaft A can only be rotated when the reversing shaft B is in one of two positions (ahead and astern) defined by the positions of the gaps cut in the circumference of a disc keyed thereto. Furthermore, the shaft B can only be rotated from its ahead to its astern position (or vice versa) when the manœuvring shaft A is in one position —the stop position. The solution when the shafts are at right angles, as in Fig. 271, is equally obvious. An indefinite number of other schemes could easily be devised to meet the requirements of different arrangements of gear.

Hand Controls.—It is essential that all the wheels and levers by means of which the engine is controlled should be grouped together so that they may be manipulated by one man in one position. In the Werkspoor Engine the arrangement of controls is particularly neat, and consists of a row of hand levers, arranged at a convenient height at the centre of the engine, and one of which is used to answer the telegraph. In the Burmeister & Wain Engines one long hand lever is

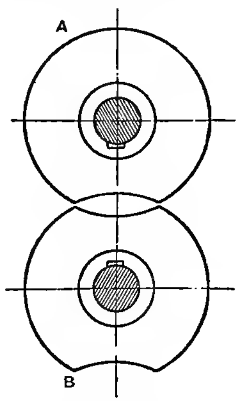


FIG. 270.

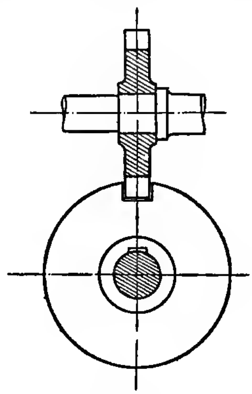


FIG. 271.

provided for the manœuvring process of switching over from air to oil, and also the regulation of the fuel supply. A separate, smaller hand lever is provided for reversing, and a wheel is fitted for emergency hand reversing. Under normal conditions the engine is therefore controlled by two levers.

In a recent Polar Diesel Marine Engine a somewhat novel departure has been made in centralising the controlling gear on the top platform. This position gives the engineer an advantage in having the fuel valves under observation. The fuel pumps are also grouped in the immediate neighbourhood.

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